9 Engine Cooling

Klaus Mollenhauer and Jochen Eitel

9.1 Internal Engine Cooling

9.1.1 The Function of Engine Cooling

9.1.1.1 Heat Balance and Heat Transport

Depending on its size, principle of operation and combustion system, a diesel engine converts up to 30–50% of the fuel energy supplied into effective brake work. Apart from conversion losses during combustion, the remaining percentage is released into the environment as heat (Fig. 9-1), predominantly with the exhaust and by the cooling system. Only a relatively small percentage reaches the environment by convection and radiation through the surface of the engine. In addition to the component heat transferred to the coolant, the heat dissipated by a cooling system also includes the heat dissipated in the lubricating oil cooler and intercooler.

Utilizing the energy loss for heating purposes and the like (see Sect. 14) requires a detailed analysis of the enthalpy content of the individual kinds of heat as well as the engine's use and type. The external cooling system (Sect. 9.2) also has to be incorporated in the analysis. Internal engine cooling essentially covers the wall heat losses that occur when energy is converted in the combustion chamber (see Sects. 1.3 and 7.2) and reaches the coolant by heat transmission. Other engine components, e.g. injection nozzles, exhaust gas turbochargers and exhaust manifolds, are often directly cooled too.

Analyzed from the perspective of energy conversion alone, engine cooling appears to waste energy. This raises the question of whether an uncooled *adiabatic engine* might not represent a worthwhile goal of development. The belief that high temperature strength and heat insulating materials had been discovered in newly developed *ceramic materials* and an adiabatic engine, one of Rudolf Diesel's basic ideas, was one step closer was widespread in the early 1980s.

The rise of engine component temperatures to approximately 1,200°C when cooling is discontinued was already pointed out in 1970. Even today, this remains an uncontrollable temperature level for reciprocating piston engines [9-1, 9-2] and is compounded by the decrease of the cylinder charge and thus the specific power at such wall temperatures when the charge loss is not compensated by supercharging or increased boost pressure. Experimental tests on an engine with an insulated combustion chamber detected a noticeable deterioration of fuel consumption instead of the expected improvement in consumption [9-3]. A strong rise of the gas-side heat transfer coefficient in the first part of combustion, thus causing more rather than less heat to reach the coolant, was demonstrated to be the reason for this (see Sect. 7.2). Engine process simulations ultimately revealed [9-4] that effective engine cooling that prevents component temperatures from rising above the level common today is one of the basic prerequisites for low nitrogen oxide emission.

Thus, one essential function of engine cooling is to lower the temperatures of the components that form the combustion chamber (piston, cylinder head and cylinder liner) far enough that they retain their strength. Limited thermal expansion of a piston must ensure that the running clearance is sufficient to prevent frictional wear between the cylinder liner and piston. Moreover, the lubricating oil must have the requisite viscosity and may not be thermally overloaded. High temperature corrosion establishes an additional temperature limit for highly thermally loaded exhaust valves (see Sect. 4.3).

In addition, engine cooling also serves to

- improve performance by better charging,
- reduce fuel consumption and exhaust emission,
- improve turbocharger compressor efficiency and, finally,
- keep engines safe and protect operators.

Engine cooling is basically divided into *liquid cooling* and *air cooling*. The type of cooling given preference depends on the engine's level of power and type of use, climatic conditions and often buyers' attitudes as well. Market demands and rationales related to use have resulted in certain concentrations of use of liquid and air-cooled engines in the past: Low and medium power high speed diesel engines for construction, agricultural and auxiliary equipment in particular, are frequently air-cooled. Air-cooled engines are limited to only just a few makes in the commercial vehicle sector and have

K. Mollenhauer (✉)
Berlin, Germany
e-mail: klamoll@aol.com

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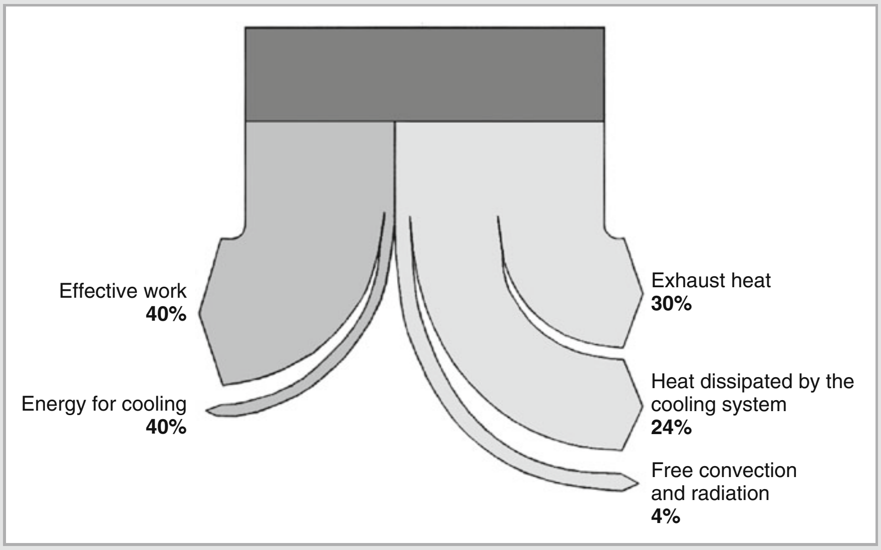

Fig. 9-1
External heat balance of a modern commercial vehicle diesel engine

largely been edged out of the car sector by liquid-cooled diesel engines in recent years.

Unlike air-cooled engines, liquid-cooled engines dissipate component heat to the environment *indirectly* rather than directly. A closed, separate cooling circuit transports the heat absorbed by the coolant in the heated zones of the cylinder head and cylinder to a coolant/air or coolant/water heat exchanger where it is released to the environment as depicted schematically in Fig. 9-2. Mathematical modeling of this heat transport process in its entirety, which consists of several processes of heat transfer and heat conduction, is extremely involved. Engine engineers normally restrict themselves to the heat transport from the working gas to the coolant, which, in simplified terms, corresponds to the transmission of heat by a flat wall. Section 7.2 already described the transfer of heat from the working gas to the surrounding combustion chamber walls.

A liquid-cooled engine's coolant is normally water or a water/ethylene glycol blend with anti-corrosion and anti-cavitation additives. The addition of ethylene glycol can lower a coolant's freezing point to −50°C. While the cooling effect of

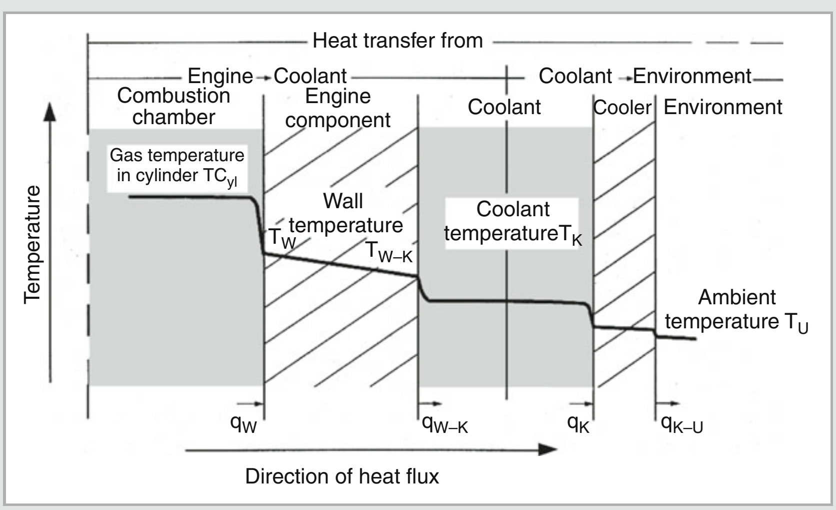

Fig. 9-2
Temperature curve and heat fluxes in engine and cooling system

water without added ethylene glycol is superior to any other coolant, pure water cooling is limited to only medium and low speed marine diesel engines and a few auxiliary equipment engines in which either the aid of external power or the engine's location make it impossible for the cooling water to freeze (For the sake of simplicity, water/ethylene glycol cooling is referred to as water cooling from here on.)

Utilized for some lower power high speed engines, *oil cooling* is a special case of liquid cooling (but is not the same as utilizing lubricating oil for piston cooling; see Sect. 8.6). The cylinder heads and cylinders in oil-cooled engines are cooled with the engine's lubricating oil so that only two resources (lubricating oil and fuel) are needed and additional coolant can be dispensed with. However, the achievable cooling capacity is limited since lubricating oil has poorer properties for cooling purposes (see Sect. 9.1.3).

9.1.1.2 Mathematical Analysis of Heat Transport

Along with model tests and experimental investigations on prototypes, a variety of CAE techniques can now be drawn on to analyze cooling water transport in an engine's heated components. Given the filigreed geometry of engine components and their cooling chambers evident in the engines presented in Sect. 17.1, the available programs based on the *finite element method* are truly optimally suited to simulate flow models as well as temperature fields and heat fluxes. They may be used to already analyze the effects of varying boundary conditions, e.g. geometry changes or parameter variations, during the design stage without consuming much time or money.

Such simulations presuppose the generation of an FEM system for an engine block's cooling water jacket as pictured in Fig. 9-3. Thus, for example, a *flow model* can serve as the basis to analyze the velocity and pressure distributions in an engine's entire coolant jacket, localize existing dead zones and optimize the coolant flow rate in the area of the valve bridge. Further, it is also now possible to use the value of the coolant-side heat transfer coefficient to obtain guide values.

Employing these values and utilizing engine process simulation, the temperature distributions inside components can be presimulated on the basis of structural models of the cylinder head and the cylinder crankcase. The temperature distribution in the cylinder head bottom of an oil-cooled engine presented in Fig. 9-4 is the result of such a simulation.

9.1.2 Water Cooling

9.1.2.1 Water Cooling Heat Transfer

Sufficient cooling of cylinder heads and cylinder liners *with coolant* is the prerequisite for effective component cooling. Dead zones must be prevented in the cooling chambers. The coolant flow must purge vapor locks that form in the highly thermally loaded zones of components. Figure 9-2 contains values that describe the heat transfer from a component to the coolant. Assuming that the heat transferred from the component wall to the coolant corresponds to the heat absorbed by the cooling system and inserting the heat transfer coefficient and the wall temperature on the coolant side, the following applies to the *heat flux density*

$$q_{WK} = q_K = h_K(T_{WK} - T_K).$$

The coolant temperature and local wall temperatures can be measured relatively easily. Determining the heat transfer

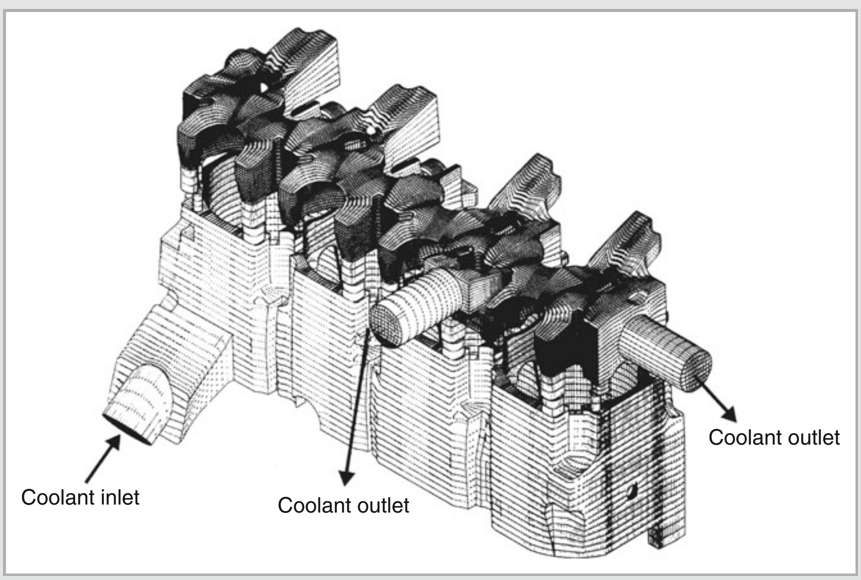

Fig. 9-3
FEM mesh of a car diesel engine's coolant jacket (V_H=1.9 l; VW)

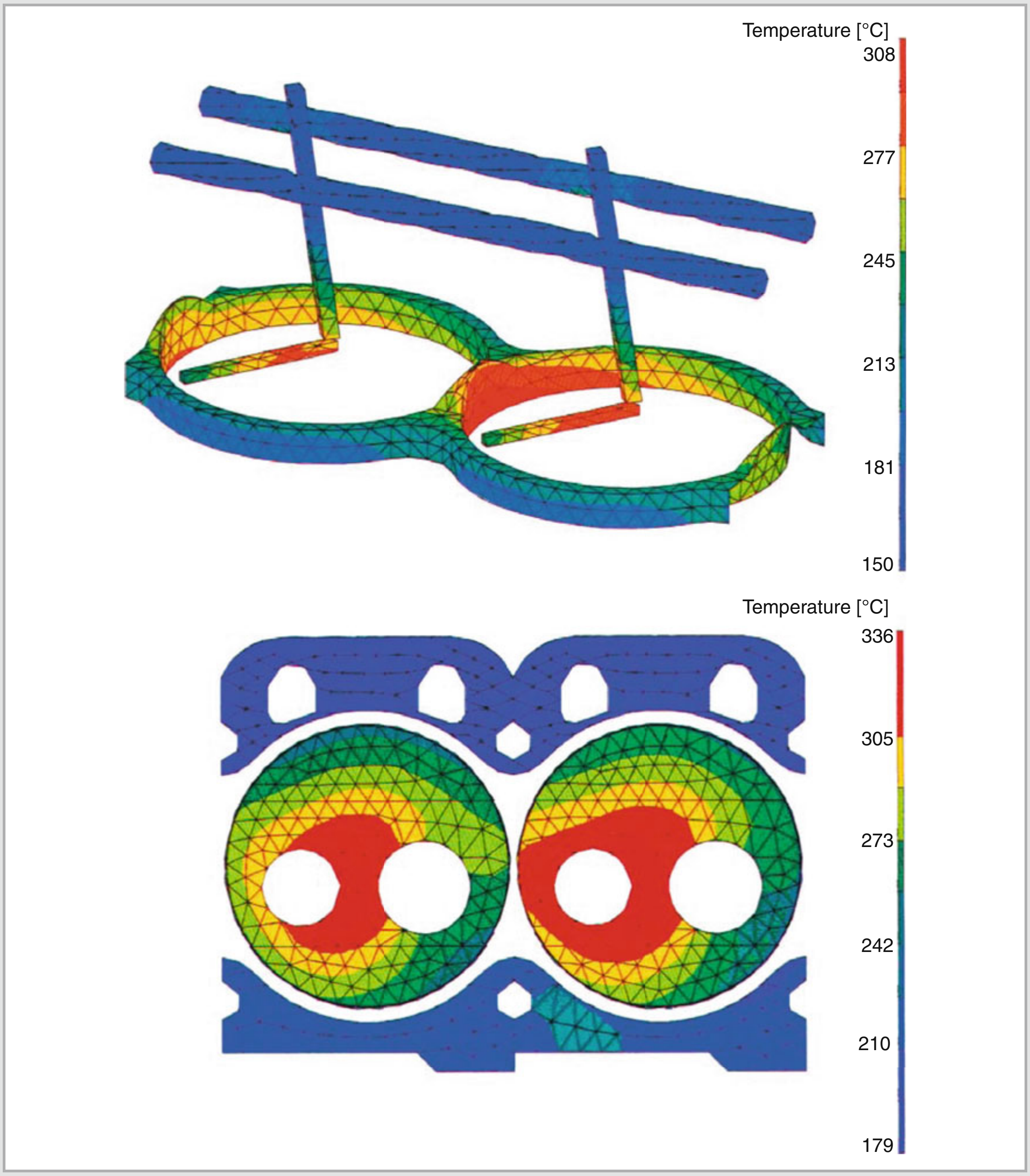

Fig. 9-4 Temperature distribution in a Deutz BF4M 2011 oil-cooled diesel engine's cylinder head bottom (*top*) and cylinder head base plate (*bottom*); see Fig. 9-10 for the cooling oil flow

coefficient for the heat transfer on the coolant side proves to be significantly more difficult. Along with the cooling water's flow velocity, density, specific heat capacity, thermal conductivity and composition, it also depends on the component's thermal load and shape as well as the flow conditions at the component. In addition, nucleate boiling and cavitation caused by engine vibrations sometimes strongly influence the heat transfer locally. Thus, altogether *very different local*

heat transfer conditions may exist in one engine. The laws of heat exchange derived from similarity theory with its parameters [9-5, 9-6] and the relationships developed from them mainly apply to geometric bodies and defined flow conditions. Hence, they can only be transferred to engines conditionally [9-7, 9-8].

Cylinder liner. Cooling water chambers of a larger engines' cylinder liners (Fig. 9-5) predominantly have a vertical flow with low flow velocities far below 1 m/s. Free convection has a more dominant influence than forced convection below 0.5 m/s. As the temperature gradient $T_{WK} - T_K$ increases and the length L within the flow decreases, free laminar convection turns into free turbulent convection and heat transfer increases. In addition, rather than the smooth tubes usually assumed, the roughness on the outside of the cylinder liners also has to be allowed for. Thus, the empirical values for the heat transfer coefficients (HTC) α_K specified in Table 9-1 and in the literature [9-9] often have to suffice.

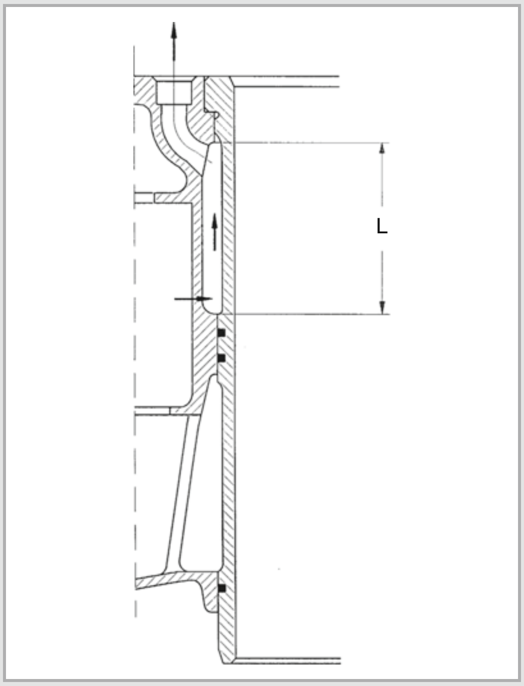

Fig. 9-5 Flow conditions in a larger water-cooled engine's wet cylinder liner

Depending on the construction of the engine block and the configuration of the cylinders (Fig. 9-6), the heat transfer to the coolant in smaller (car) diesel engines (Sect. 17.1) may be modeled as *aligned tube bundles in a cross flow* [9-10]. This was demonstrated by comparisons of engine measurements and simulations of both real engines and engine component models and with water and oil as coolant. In addition, taking the coolant velocity c_0 in the empty port cross section and the proportion of the flow port that is a cavity ψ as the starting point and employing the values specified in Fig. 9-6, the velocity c_ψ is calculated as follows:

$$c_\psi = c_0/\psi = c_0/[1 - \pi D'/(4H)].$$

Where $a = S_1/D'$ and $b = S_2/D'$, the arrangement factor f_A is determined as

$$\mathrm{f_A} = 1 + \{[0.7(\mathrm{b/a}) - 0.21]/\psi^{1.5}[(\mathrm{b/a}) + 0.7]^2\}$$

With the outer diameter D' of the cylinder in the flow and the characteristic "overflow length" $L = \pi \cdot D'/2$ and using c_ψ as the velocity term, the following ensues for the Reynolds number:

$$\mathrm{Re}_\psi = c_\psi \cdot \mathrm{L}/\nu = c_\psi \cdot \pi \mathrm{D}'/2\nu$$

Thus, when the Prandtl number *Pr* is known, the independent Nusselt number Nu^* can be determined (Fig. 9-7) as

$$Nu^* = Nu/f_R\,,$$

or when the tube bank coefficient

$$f_\mathrm{R} = [1 + (z-1)f_\mathrm{A}]/z,$$

incorporates the number of cylinder barrels z in the flow, the Nusselt number $Nu = h_K L/\lambda$ and thus the *mean heat transfer* h_K can be determined for the heat transfer in the cylinder barrels in the flow (see Table 9-2 for the physical properties).

In highly loaded water-cooled engines, the piston's impact against the cylinder liner when it changes contact with the cylinder liner can cause cavitation in the cylinder liner on the coolant side [9-11]. This increases the heat transfer coefficient locally by as much as tenfold and speeds up the destruction of materials induced by the cavitation attack (see Sect. 7.1).

Table 9-1 Guide values for coolant-side heat transfer coefficients

Type of heat exchange	h_K in W/(m$^2 \cdot$ K)		
	Water	Water/ethylene glycol 50%/50%	Oil
Free convection	400. . . 2,000	300. . .1,500	
Forced convection	1,000. . . 4,000	750. . .3,000	300. . .1,000
Nucleate boiling	2,000. . .10,000	1,500. . .7,500	

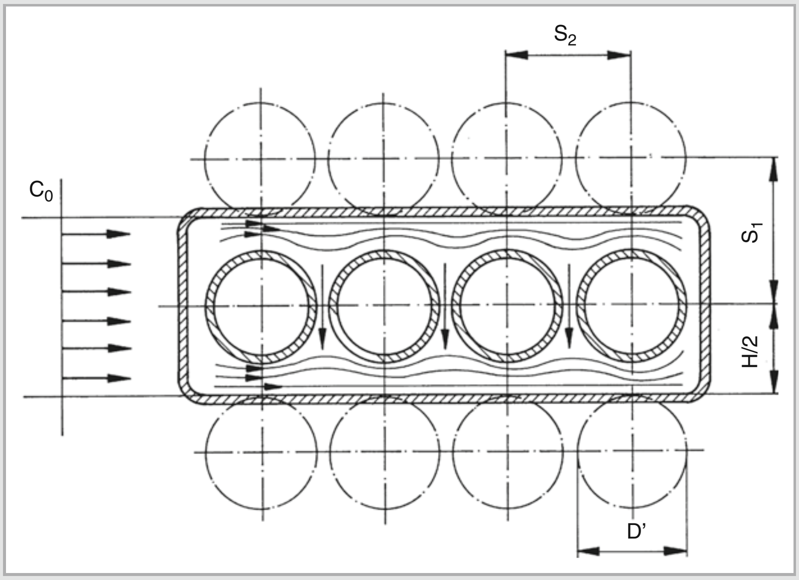

Fig. 9-6
Flow conditions and cylinder configuration for vehicle engines

Cylinder head. Given their complex arrangement, every law-like approach to the HTC, e.g. relationships for "plates in a transverse flow" for cylinder head bottoms, falls short for most cylinder heads too, particularly because a proper flow rarely develops. Moreover, since there are only low flow velocities in a cylinder head and the thermal load is higher, the heat transfer is largely determined by intensive nucleate boiling [9-8] with local heat transfer coefficients h_K of up to 20,000 W/(m^2 K) (see Table 9-1 and Sect. 9.1.2.3).

Accordingly, clear flow conditions only exist for bore-cooled engine components (Fig. 9-8) where there is a turbulent pipe flow with relatively large flow velocities induced by the forced flow in narrow pipes. Nearly every surface temperature desired may be obtained by varying the bores' spacing and diameters and their distance to the surface and the coolant flow rate. Hence, the thickness of the main walls decisive for mechanical stress may be freely selected [9-12]. Thus, *bore cooling* takes up the design principle of separating thermal and mechanical loading (see Sect. 7.1).

9.1.2.2 High Temperature Cooling

As the name indicates, the significantly *higher temperature level of the coolant* distinguishes high temperature cooling from a conventional cooling system. High temperature cooling aims for temperatures of up to 150°C on the engine's coolant outlet side in the part load range. Naturally, this results in a corresponding rise of component temperatures. However, suitable control of the coolant temperature must ensure that the maximum permissible component temperatures are not exceeded in any of the engine's load points. A component temperature-regulated cooling concept is described in [9-13]. Since the vapor pressure of water at 150°C is just under 5 bar, engines with high temperature cooling cannot use a conventional coolant unless the intention is to put the entire cooling system under high pressure corresponding to the boiling pressure. Experience has shown that the hermetic sealing this requires is unachievable. Thus, the minutest leaks cause an engine to "mist".

Aircraft engines operated with high temperature cooling in the past used pure ethylene glycol as coolant. However, pure ethylene glycol reduces the heat transfer coefficient to a fifth of that of pure water. The *advantage of high temperature cooling* is a perceptible drop in fuel consumption in the engine's lower part load region. The comparatively higher lubricating oil temperature and thus lower oil viscosity is the reason for this and reduces an engine's hydrodynamic frictional losses [9-14]. Despite the advantages in consumption and intensive research primarily in the 1970s, high temperature cooling has still not become widespread in high speed vehicle and industrial engines because a suitable cooling medium that meets the diverse requirements imposed on a coolant is lacking. Described in Sect. 9.1.3, oil cooling in which the maximum coolant temperature at the engine outlet is 130°C represents an important step toward high temperature cooling.

9.1.2.3 Evaporation Cooling

Basics

Evaporation cooling is based on the physical principle of cooling by latent heat and the inherent regulation of temperature

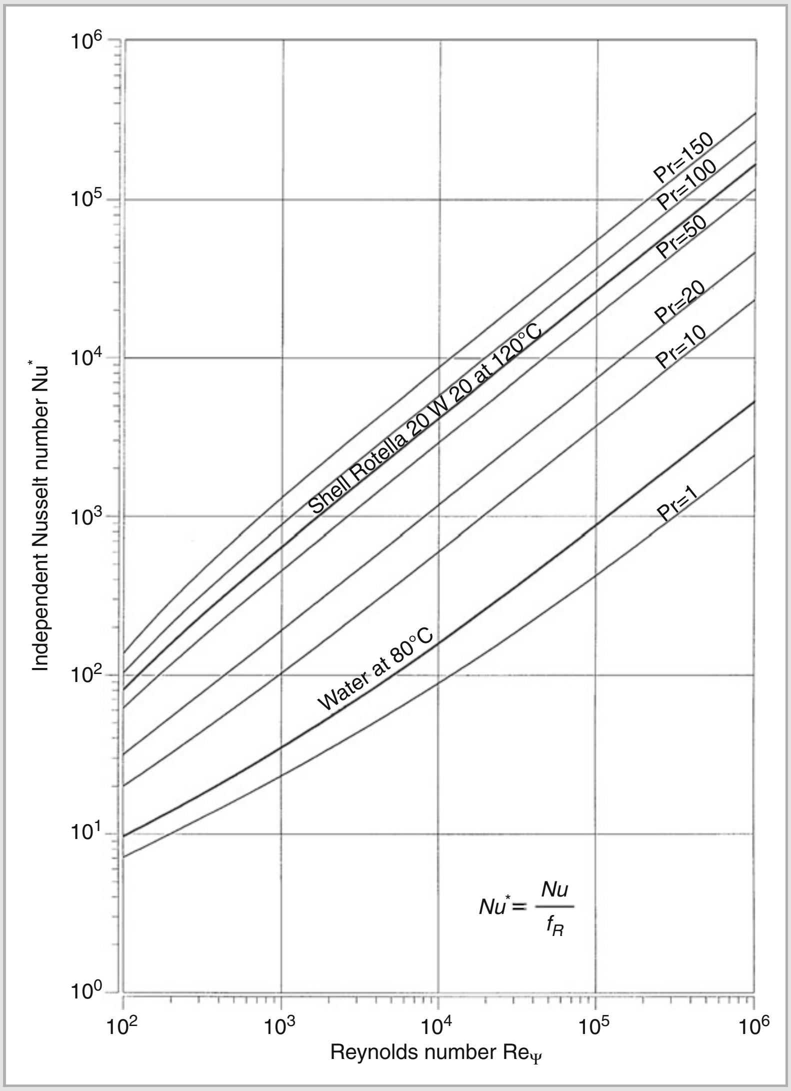

Fig. 9-7
Nusselt number as a function of the Reynolds number and Prandtl number for coolant-side heat transfer in integrated cylinder liners in vehicle engines

Table 9-2 Physical properties of air and water

			Air at 60°C	Water at 95°C	Ratio of air to water
Density	ρ	kg/m^3	1.045	961.70	1 : 920
Spec. heat capacity	c_p	kJ/(kg · K)	1.009	4.21	1 : 4.2
Thermal conductivity	λ	W/(m · K)	$28.94 \cdot 10^{-3}$	$675.30 \cdot 10^{-3}$	1 : 23
Kinematic viscosity	ν	m^2/s	$18.90 \cdot 10^{-6}$	$0.31 \cdot 10^{-6}$	61 : 1
Thermal diffusivity	a	m^2/s	$27.40 \cdot 10^{-6}$	$0.17 \cdot 10^{-6}$	161 : 1

connected with it. The cooling liquid in the components being cooled is heated to boiling temperature so that *nucleate boiling* causes heat to transfer to the coolant without having to force the flow through the engine's cooling chambers: When the thermal load of the component is higher, the temperatures on the surface on the cooling side rise so that an overheated state is reached in the boundary layer near the wall although the cooling liquid's mean temperature is below its saturation

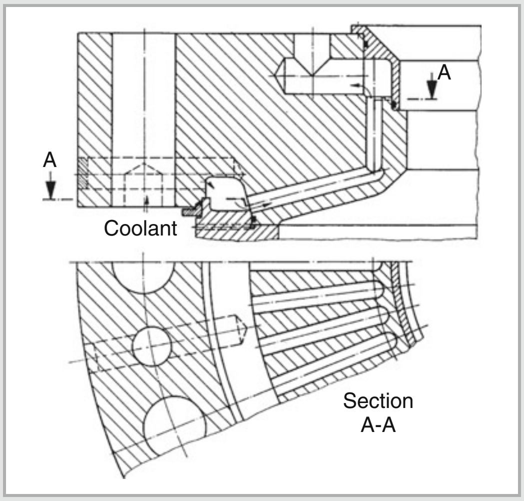

Fig. 9-8 Bore cooling for a cylinder head of a large two-stroke diesel engine (MAN B&W) based on [9-8]

temperature. The vapor locks produced by this are swept along by the coolant flow, thus causing implosions in the proximity of the wall and an intensive pulsation flow as a result of the continuing vapor lock. This increases as the thermal load increases and thus the heat transfer coefficient HTC increases too. An experimentally corroborated relationship for the HTC published in [9-10, 9-15] incorporates the simultaneous occurrence of nucleate boiling and convection.

Nucleate boiling occurs more or less strongly in every highly thermally loaded component, even in the top region of cylinder liners. The strong rise of the HTC connected with this translates into thermal self-protection for the engine to a certain extent since the increased heat transfer only causes the wall temperature on the coolant side to increase by approximately 10–20 K. Nonetheless, the larger temperature gradient in a component when nucleate boiling is local can cause stronger deformations and stresses, especially during dynamic load operation.

Closed System Evaporation Cooling

The advantages of this type of cooling over forced circulation cooling include a substantially more even temperature distribution in the component while temperature fluctuations in the coolant as a function of load are only slight. Since the coolant's boiling temperature depends on the vapor pressure and thus the system pressure, programmed pressure control can keep the components' temperature level roughly constant at differing engine loads. Thus, as with high temperature cooling (see Sect. 9.1.2.2), lower fuel consumption is attained in part load operation as a result of the higher component temperatures that persist in the normal range even at full load. The vaporized coolant liquefies in the schematic of the closed circuit evaporation system presented in Fig. 9-9. An electrically driven coolant pump constantly circulates the coolant, which exists in both a liquid and gaseous state, through the engine cooling chambers, through the bypass line when the engine is cold and through the heat exchanger when the engine is warm. The fluid/vapor separator ensures that primarily vapor is conducted to the heat exchanger. Given the water-based coolant's high heat of evaporation, its volumetric flow is only approximately 1–3% of its value in forced circulation cooling. Thus, relatively small pumps with greatly reduced power consumption may be used to facilitate lower fuel consumption.

To prevent cavitation in the pump inlet, the coolant suctioned in must be sufficiently undercooled by designing the

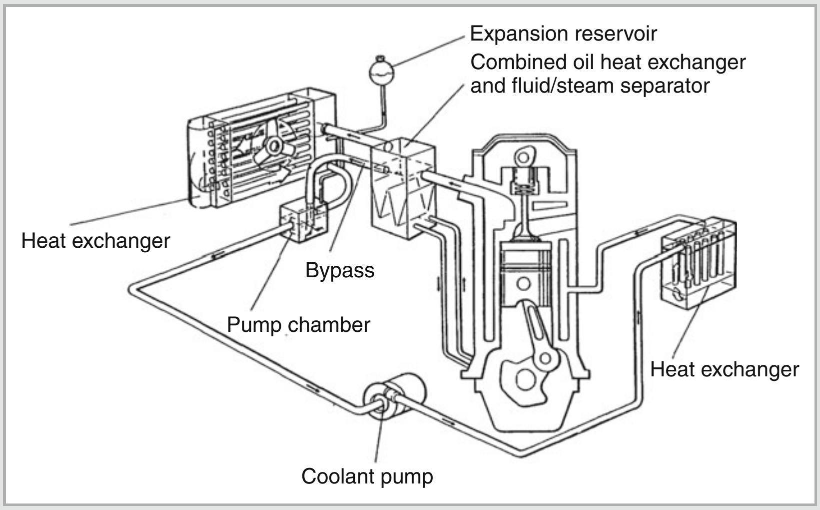

Fig. 9-9
Schematic of evaporation cooling (VW)

vehicle heat exchanger and the coolant flow appropriately. The flow of vapor into the heat exchanger is produced by the partial pressure gradient opposite the engine cooling chambers.

In thermal equilibrium, a common ethylene glycol/water blend with a content of 50/50% by volume produces system-independent boiling temperatures of 105–120°C at a differential pressure in the system of 100–300 mbar, i.e. significantly lower than the liquid circulation cooling common in cars. This reduces the requirements on components' compressive strength. The achievable cuts in consumption are as much as 5% for gasoline engines but no more than 3% for diesel engines because of their lack of thermal dethrottling [9-16, 9-17].

It is problematic that the usually additized ethylene glycol/water blend is not azeotropic, i.e. the low boiling water fraction in the engine's cooling chamber distills and thus causes the glycol concentration to increase. This entails a steady rise of the boiling temperature and a constant feed of undercooled cooling liquid to compensate. On the other hand, the glycol fraction in the heat exchanger and thus the antifreeze protection are reduced. Suitable alternatives for the cooling liquid have not been found yet.

Moreover, the continued lack of solutions to challenges in the design of cooling chambers, e.g. additional space requirements for the water separator, expansion reservoir, larger pipe cross sections, etc., explains the failure thus far to develop a state-of-the-art engine with evaporation cooling to the stage of production despite its advantages, particularly since it does not generate any cost advantages over conventional cooling in terms of the overall technical work required.

Open System Evaporation Cooling

Used earlier even in Europe for small, rugged single-cylinder diesel engines of low power, this cooling system has been implemented as a horizontal single-cylinder diesel engine millions of times, e.g. in China, to motorize so-called *walking tractors,* which, among other things, relieve farmers or construction workers from heavy physical labor. The evaporated cooling water can be replenished with simple untreated water.

9.1.3 Oil Cooling

Oil cooling alters both the thermal stress of components and the running properties of an engine. Oil neither freezes nor boils in the relevant temperature range of −50°C to 150°C. Oil-cooled engines (Fig. 9-10) do not experience corrosion

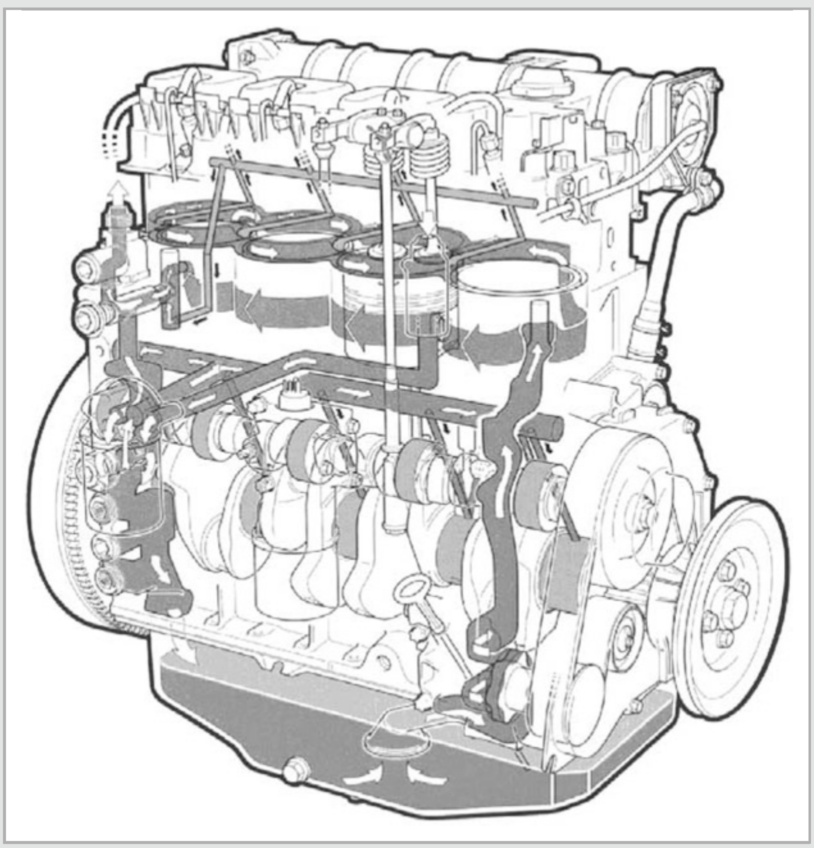

Fig. 9-10
Schematic of a Deutz BF-4 M oil-cooled diesel engine's lubricating and cooling oil flow [9-19]

and cavitation problems. Cooling oil's working temperature of 100–130°C is significantly higher than the usual cooling water temperatures. This results in a correspondingly higher component temperature level, which, in turn, leads to slightly less component heat being eliminated by the coolant and the heat capacity of the exhaust gas flow being somewhat larger than in conventional water-cooled engines. With its low specific fuel consumption in the lower part load region, an oil-cooled engine comes quite close to a "high temperature cooled" engine.

Oil heated to 130°C has a density approximately 15% lower, a specific heat capacity roughly only half as large, thermal conductivity approximately only a fifth as high and a dynamic viscosity 10–30 times greater than water at 95°C. Given the coolant's greater viscosity, particular attention must be devoted to the design of the cross section of cooling chambers and cooling oil bores for oil-cooled engines.

The heat transfer coefficient in oil cooling reaches only 25–30% of the value in water-cooled engines (see Table 9-1). The exponent of the velocity in the correlation $h_K \sim c^n$ averages 0.3. Given the moderate heat transfer coefficient on the coolant-side, an oil-cooled engine naturally cannot attain the same specific power as a water-cooled engine. A supercharged, oil-cooled engine's maximum specific power is 21 kW/l. However, the less intensive heat transfer and the higher coolant temperatures have the advantage of lower temperature gradients in the component and thus significantly lower material stress. Compared to water cooling, oil cooling is *gentle cooling*.

The new 2011 series of oil-cooled diesel engines [9-18] has been presented as a further development of the 1011 series [9-19] (see also Sect. 18.2). Primarily designed for light tractors, construction equipment and auxiliary equipment, these direct injection engines with a power of up to 65 kW at a nominal speed of 2,800 rpm are distinguished by an "open deck" oil-cooled cylinder crankcase with cast-in cylinder liners and a cylinder head with block bore cooling, which is also oil-cooled. The cylinder's oil flows in the longitudinal direction of the engine. All the coolant delivered by the oil pump initially flows around the cylinder walls (Fig. 9-10). The flow cross sections in the crankcase were varied from cylinder to cylinder to ensure that the flow through the intermediate cylinder walls is also adequate. The cylinder head bottom's cooling chamber geometry is a mirror image of the cylinder crankcase and cooling oil flows through it in the opposite direction (see Fig. 9-4).

The cylinder head gasket separates the crankcase and cylinder head cooling chambers. The cooling oil from the crankcase flows through bores located in it into the cylinder head's cooling gallery. Upwardly inclined and interconnected bores meter every cylinder unit's outflow of oil. Two V-shaped bores arranged in the cylinder head above the last cylinder on the side of the fan connect the cooling gallery in the cylinder head bottom with the return gallery. All in all, this reduces the resistance to flow in the cylinder head, increases the flow rate and improves the cylinder head's cooling by uniformly distributing the cooling liquid. The experiences acquired with oil-cooled engines have demonstrated that deposits do not form in the cooling chambers, viscosity properties do not deteriorate and oil oxidation does not increase.

9.1.4 Air Cooling

9.1.4.1 Historical Review

The idea of air cooling – dissipating the heat of an engine's components directly to the ambient air – is as old as the internal combustion engine itself. The Frenchman de Bisschop already introduced an air-cooled internal combustion engine in 1871. Lenoir's single cylinder gas engine, which operated according to the atmospheric principle [9-20], had longitudinal fins cast on the working cylinder, which conducted the cooling energy to the environment by free convection.

The meteoric development of the aviation industry after Bleriot's flight across the English Channel in 1909 also included the development of an air-cooled aircraft engine and was marked by the following milestones: The development of aluminum alloys (1915), the introduction of light alloy cylinder heads (1920) and research on the physical correlations of heat dissipated by cooling fins [9-21], their optimal design [9-22, 9-23] and the influence of the cooling airflow routing.

Apart from the aircraft gasoline engines that attained powers per unit piston area of approximately 0.5 kW/cm^2 in 1944, many firms throughout the world saw themselves induced in the 1930s to work on the development of aircraft diesel engines to counter the danger of altitude induced misfires and achieve higher efficiencies. Out of a total of twenty-five projects, twelve engines were air-cooled, yet only Junkers' Jumo 205 water-cooled opposed piston engine was produced in larger quantities. Air-cooled vehicle diesel engines were launched on the market at approximately the same time. In 1927, Austro-Daimler introduced the first high speed diesel engine, a four-cylinder inline engine with 11 kW of power.

Water-cooled engine systems' susceptibility to relatively major breakdowns under extreme climatic conditions brought the industry orders for developments of air-cooled commercial vehicle and tank diesel engines during the Second World War. Building upon the results obtained, air-cooled diesel engines were developed and implemented in commercial vehicles and agricultural and construction equipment as of the 1950s [9-24–9-26]. In those days, a few engine firms deliberately only offered air-cooled diesel engines: Klöckner-Humboldt-Deutz (now Deutz AG) has been the global market leader since the 1950s. Around 1980, air-cooled diesel engines could be found in 80% of the construction equipment made in Germany.

Initially, it was possible to counter the increase in thermal component loading with a trend toward higher power density by switching to direct injection and employing aluminum

alloys with higher high temperature strength. Yet, the number of air-cooled diesel engines produced has been declining since the mid 1980s. Small industrial diesel engines with approximately 15 kW of power are still almost solely air-cooled because of the cost advantage of the integrated cooling system. Even today, air-cooled diesel engines with up to approximately 100 kW of power are still preferred for construction equipment and auxiliary equipment. Both types of cooling exist side by side in the higher power range of up to approximately 400 kW [9-27, 9-28].

9.1.4.2 Heat Exchange from Components to the Cooling Air

Heat Transfer and Cooling Surface Design

The heat transfer coefficient h, temperature gradient ΔT_K and heat exchanging surface A determine the transferrable heat (see Sect. 9.1.2). The heat transfer coefficient (HTC) is a function of the flow velocity c and the material properties (coefficient of thermal conductivity λ, kinematic viscosity v and thermal diffusivity a). The comparison of physical properties in Table 9.2 indicates the different heat transfer conditions for air and liquid cooling.

With the aid of similarity theory and its parameters, i.e. the Nusselt number $Nu = h \cdot D/\lambda$, Reynolds number $Re = c \cdot D/v$ and Prandtl number $Pr = v/a$, the heat transfer in a turbulent flow tube with an internal diameter D can be described by the following power equation ($10^4 < Re < 10^5$):

$$N_u = 0.024 \cdot Re^{4/5} \cdot Pr^{1/3}.$$

Retaining the tube diameters and flow velocities, the following ensues for the relationship of the heat transfer coefficients for air and water (index "Lu" and "Wa"):

$$h_{Lu}/h_{Wa} = (\lambda_{Lu}/\lambda_{Wa}) \cdot (v_{Lu}/v_{Wa})^{-7/15} \cdot (h_{La}/h_{Wa})^{-1/3},$$

and the following after inserting the physical properties (Table 9.2):

$$h_{Lu}/h_{Wa} = 1870.$$

Thus, under identical conditions, the heat transfer coefficient for water cooling is approximately 900 times greater than for air cooling. However, its significantly lower density than water's allows considerably greater air velocities and thus an eight to tenfold increase of the heat transfer. Despite the improvement this makes possible, according to a relation of

$$h_{Lu}/h_{Wa} = 1/(110...60)$$

the same cooling capacity as for water cooling can only be dissipated by larger temperature gradients between a component and coolant and by giving the components fins and thus enlarged surfaces. With the same cooling capacity and an empirical ratio of temperature differences of

$$(T_{WK} - T_K)_{Lu}/(T_{WK} - T_K)_{Wa} \approx 2...4$$

the heat dissipating surface on the cooling side must be enlarged to

$$A_{Lu}/A_{Wa} \approx 15...55.$$

For *straight fins with a rectangular cross section*, a fin height h, width b and space width s and a coefficient of thermal conductivity λ_R, the following applies to the cooling energy flow q_K at a wall temperature T_{WK} in the fin base and a cooling air temperature T_K:

$$q_K = h_K(T_{WK} - T_K)[(2h + s)/(b + s)] \cdot \eta_R$$

with the fin efficiency factor

$$\eta_R = \tanh(h\sqrt{2h_K/\lambda_R b})/(h\sqrt{2h_K/\lambda_R b}),$$

which compares the heat flux actually released by one cooling fin with that a fin with constant surface temperature would transfer, i.e. with infinitely large thermal conductivity. As indicated in Fig. 9-11, the fin height necessary for heat exchange and thus the importance of finning decreases rapidly as the heat transfer coefficient grows larger. Therefore, the cooling chamber walls of water-cooled engines remained unfinned. For the limit case where fin height and fin thickness approach zero, the equation for the heat flux density is simplified to the form that applies to a flat smooth wall $q_K = h_K (T_{WK} - T_K)$.

Knowledge of not only the local component temperature but also the value of the heat transfer coefficient is the prerequisite for a sufficiently precise determination of the cooling energy flow. In principle, the different equations for the heat transfer coefficient can be traced back to one of the approaches of [9-8]

- heat transfer in a turbulent flow channel or
- heat transfer in a body within a flow.

For the finned channel of a fully encased finned tube, the first approach to channel flow can revert to a multitude of validated heat transfer equations for turbulent pipe flow inserting the hydraulic diameter [9-29]. The second approach is based on *Krischer* and *Kast's* studies [9-30, 9-31] and corresponds to an unencased finned tube. The mathematical treatment of heat transfer applied to an air-cooled engine is described in [9-27]. Experimentally obtained heat transfer coefficients must be reverted to in cases when the flow conditions in the finning are not known precisely enough [9-32].

The *enlargement of the exothermic surface* by means of finning can be achieved both by greater fin height and a larger number of fins. However, the two options do not inevitably lead to the same increase of heat flux density when the enlargement of their surfaces is identical. Beyond a certain extent, the increase of fin height does not cause any heat dissipation since the temperature of the fin tip already approaches that of the cooling air. As the comparison of aluminum finning with gray cast iron finning in Fig. 9-12 reveals, the thermal conductivity of the fin material decisively determines the temperature curve between the fin base and fin tip.

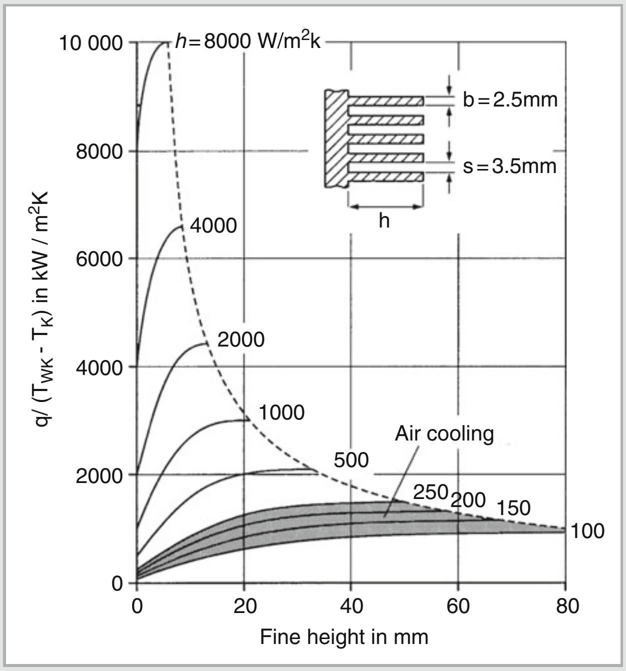

Fig. 9-11
Influence of fin height and the heat transfer coefficient on the heat flux relative to the temperature gradient for gray cast iron finning with λ_R=58 W/(m K)

Likewise, the surface can only be enlarged to a certain extent by increasing the number of fins and reducing the space width. This is defined by the space width at which the flow turns turbulent immediately before the adjacent boundary films coalesce. The minimum, still economical space width is approximately 1.2 mm [9-33]. Such small space widths are at the limit of manufacturability and only found in the machined steel cylinder barrels of high performance air-cooled engines.

Fin heights of up to 70 mm with minimum space widths of 3.5 mm at the fin base and fin thicknesses of 3–2 mm from base to tip are cost effective and also perfectly feasible for aluminum cylinder heads produced by permanent mold casting. Given their lower thermal conductivity, gray cast iron cylinder barrels are produced by sand casting with fin heights of not more than 35 mm and space widths of 3 mm at the fin base and fin thicknesses of 2.5–1.5 mm from base to tip. Machining of the type in aircraft engine manufacturing must be turned to for smaller fin thicknesses and space widths.

When the mean heat transfer coefficient is h_K = 250 W/(m^2K), the maximum heat dissipated in a cylinder barrel with gray cast iron fins is estimated to be approximately $0.5 \cdot 10^3$ kW/m; in a cylinder barrel with machined Alfin fins it is estimated to be approximately $0.2 \cdot 10^3$ kW/m^2. By comparison, water cooled highly thermally loaded components can dissipate up to $5 \cdot 10^3$ kW/m^2 of heat by locally occurring nucleate boiling.

Cooling Airflow Routing and Heat Transfer

Cooling airflow routing influences the heat transfer and thus the temperature distribution at a component. The cooling air flowing through a finned cylinder barrel is initially baffled at the point of incidence and then flows around the cylinder barrel until it breaks away laterally in the height of the meridian section so that a dead zone forms behind the cylinder. This causes the highest wall temperature to appear on the back of the cylinder and the lowest on the inflow side (Fig. 9-13). Air cowling by encasing the finned cylinder barrel is required to prevent different thermal expansions from warping the cylinder (Fig. 9-14). This may only be dispensed with in engines with small bore diameters and low power outputs per displacement, e.g. motorcycle engines.

The guide plates are intended to ensure cooling air passes through the fin channels without dead zones. By virtue of the principle, the absorption of the heat from the cooling air in the finned channel makes it impossible to attain a fully uniform temperature distribution in the circumference when the inflow is transverse. Experience has shown that, as a function of the heating of the cooling air by ΔT_{KL}, the following applies to the maximum temperature deviation in the circumference $\Delta T_{\mathrm{Umfang}}$ [9-34];

$$\Delta T_{Umfang}/\Delta T_{KL} \approx 0.8.$$

Consequently, the heating of the cooling air is limited to 50 K for a maximum permissible temperature deviation of 40 K.

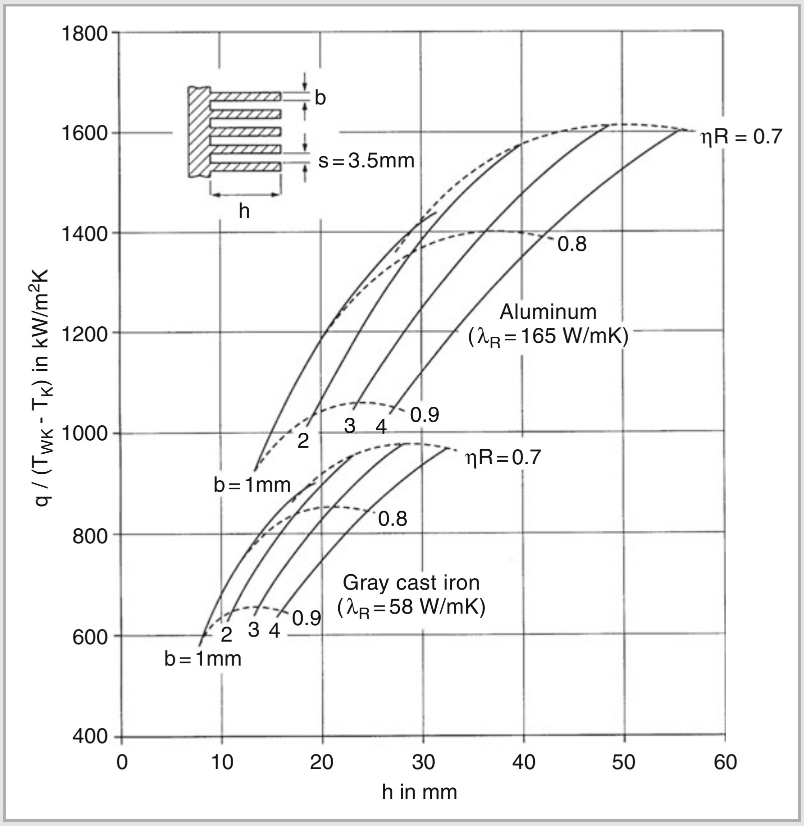

Fig. 9-12
Influence of fin height h and the fin's thermal conductivity λ_R on the heat flux relative to the temperature gradient with a constant heat transfer coefficient h_K=150 W/(m^2 K)

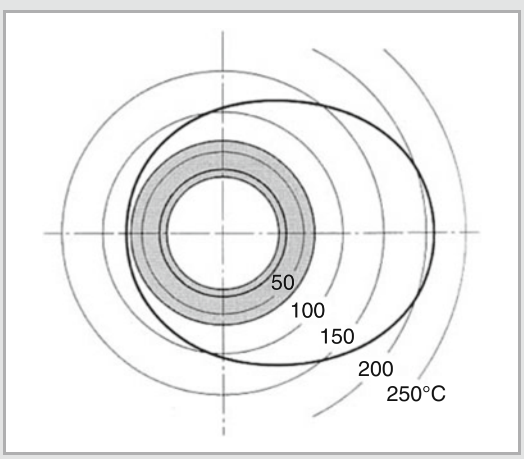

Fig. 9-13 Temperature curve for a cross-flow, unenclosed cylinder barrel (flow from the *left*).

9.1.4.3 Design Features of Air-cooled Engines

Overall Engine Design

The most striking design features of air-cooled engines are their single cylinders and integrated cooling systems. Since aluminum and cast iron have very different thermal expansion, the virtually universal use of aluminum cylinder heads (more uniform temperature distribution in the cylinder head bottom and maximum dissipation of heat to the cooling fins) and the requirement of unimpeded thermal expansion of components inevitably necessitate *single cylinder construction*. Hence, air-cooled engines are downright predestined for the modular principle with a large number of identical parts and resultant cost advantages for manufacturing and spare parts management, particularly for small and medium quantities [9-36]. Further, a modular engine allows easy maintenance around the cylinder head, cylinder barrel and piston without having to remove the engine and disassemble the oil pan.

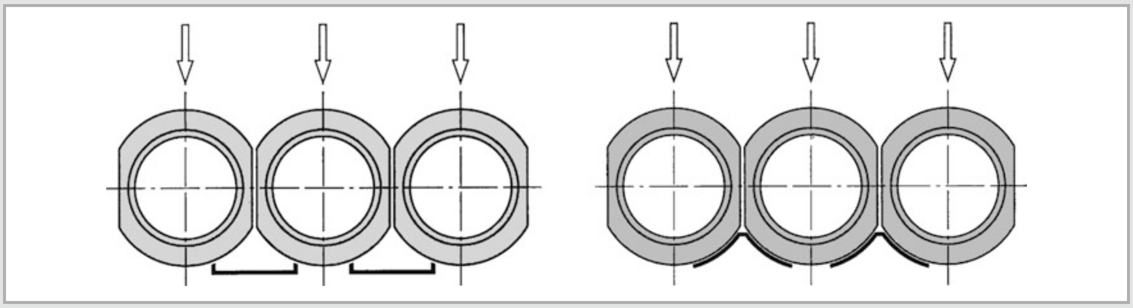

Fig. 9-14 Cooling airflow routing: Argus shroud (*left*); shroud abutting the finned tube (*right*) [9-35]

In most cases, an air-cooled diesel engine's cylinder unit is equipped with *four tension bolts* that connect the cylinder head with the crankcase, brace the cylinder barrel between the two and transfer the gas forces directly to the crankcase. Tension bolts are slender, highly elastic expansion bolts that reduce the alternating loads in the tensile stress areas of the threads and limit the increase of force resulting from the greater thermal expansion of the cylinder head and cylinder barrel than of the significantly cooler bolts.

Significant for the overall length of an engine, the design of the cooling fins and not the spacing of the crankshaft bearings determines the cylinder clearance in air-cooled inline engines. Relative to the cylinder bore, this is normally 1.35–1.45 and only 1.275 in the extreme case (KHD engine model B/FL 913).

A comparison of the installation dimensions of air and water-cooled engines including the latter's external cooling system reveals that an air-cooled engine with an integrated cooling system is an exceptionally *compact engine* despite its greater cylinder clearance. This is especially true of the V engine in which the cooling fan is mounted between the cylinder banks, thus saving space.

Crankcase

Together with the cylinder barrel, the crankcase absorbs the load transfer from the cylinder head to the crankshaft assembly with the crankshaft bearing and hence requires high inherent stability for smooth piston travel, even when the crankcase is simultaneously one of the vehicle's bearing elements, e.g. in tractors. A comparison of the cross sections of the crankcases of an air-cooled and a water-cooled engine with identical main dimensions important for their flexural and torsional rigidity in Fig. 9-15 reveals that an air-cooled engine's crankcase must be designed with greater care because of the significantly smaller overall height due to the absence of a water reservoir and the resultant poorer

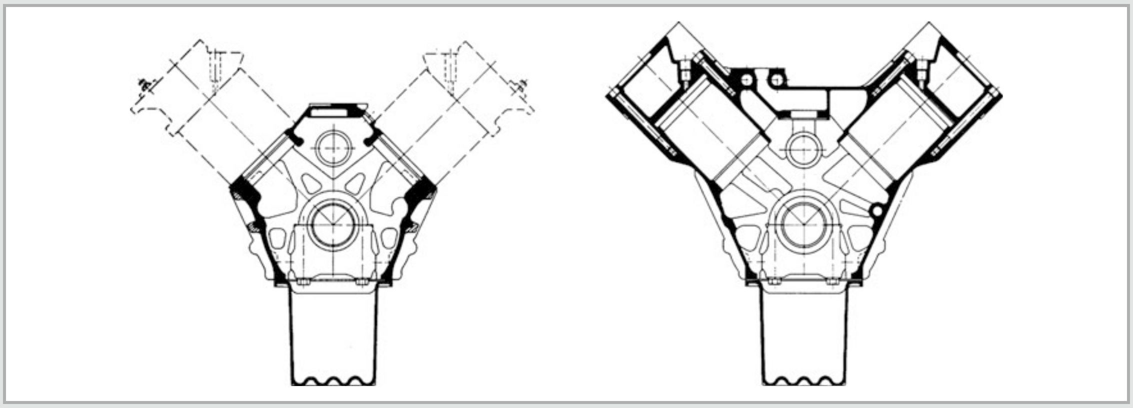

Fig. 9-15 Cross section crucial for the inherent stability of an air-cooled engine with a cylinder barrel and single cylinder head (*left*) and a water-cooled engine with a block cylinder head (*right*)

conditions for strength. Effective and proven measures that provide an air-cooled engine's crankcase high inherent stability are:

– side housing walls that are dropped far under the crankshaft axis and, if possible, curved,
– continuous bracing ribs in the side and cross housing walls and a sturdy, broad oil pan flange,
– a relatively thick, only slightly perforated supporting cylinder base deck,
– a cast oil pan instead of a sheet metal oil pan and
– cross bolting of the main bearing cover with the housing walls in V engines [9-37].

Air-cooled engines are also frequently constructed in a tunnel design for reasons of rigidity [9-27].

Cylinder Barrel

Constructed as a cylinder liner with cooling fins, the cylinder barrel is usually made of gray cast iron in one piece. Sand casting remains economical even for larger quantities. The walls in the upper and lower barrel region are usually designed somewhat thicker and the uppermost cylinder barrel zone provided with cooling fins closed along the circumference to *minimize liner deformations* caused by the bolting forces and the in-cylinder pressure. For reasons of strength and cooling, engines with high power outputs per liter displacement must switch to cast steel barrels and the filigreed body of the cooling fins must be machined.

The composite casting process (Fig. 9-16) known as the Alfin process [9-38] in which an approximately 0.03 mm thick intermediate metal layer ensures a gap-free connection between a steel barrel and its finned aluminum jacket is and has only been used in aircraft engine manufacturing with the exception of a few military applications. It is also employed for diesel engines in the commercial sector, utilizing gray cast iron instead of steel. Light alloy cylinder barrels with appropriately finished barrel surfaces are only used for gasoline engines [9-39, 9-40].

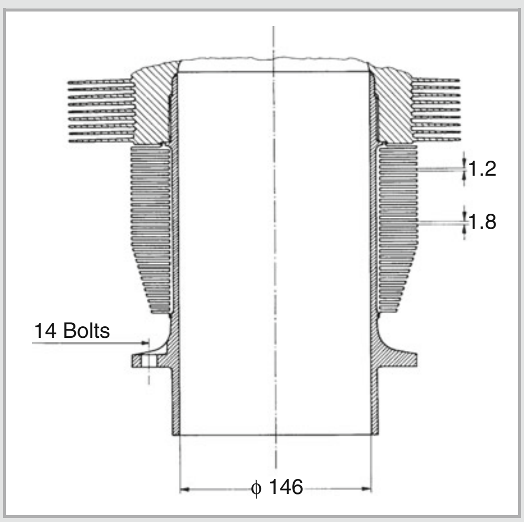

Fig. 9-16 Alfin cylinder barrel of an air-cooled diesel engine (Teledyne Continental engine) [9-28]

Cylinder Head

Its diversity of functions and alternating mechanical and thermal stresses make the cylinder head the most complex component of an engine. A cylinder head must have *high inherent stability* to transfer the gas forces acting on it to the crankcase and simultaneously ensure that the connection to the cylinder barrel is gastight. An air-cooled cylinder head not only has to house the gas ports, the injection nozzle, possibly a prechamber and the cylinder head bolts but also the cooling fins and flow cross sections necessary for the cooling air. This is a difficult task considering that a direct injection diesel engine, for example, requires a fin area 30–35 times larger than the piston area to cool the cylinder head bottom and the region of the exhaust port. In addition, the cooling fins must be designed so that the maximum cylinder head temperature between the valves remains limited and large temperature differences are avoided to prevent high thermal stresses in the cylinder head bottom. Normally, only *aluminum* cylinder heads can satisfy these requirements. Their high thermal conductivity facilitates the distribution of heat in the cylinder head bottom and allows cost effective manufacturing of thin, high cooling fins.

Such cylinder heads fundamentally require *valve seat inserts*, which are usually centrifugally cast and shrink fit. The transverse arrangement of the valves to the crankshaft axis allows a better cooling fin design in the valve region than a parallel arrangement as well as larger flow cross sections for the cooling air. However, the combustion system must allow the valves to incline strongly (gasoline engines). When they are arranged parallel to the crankshaft, the valves should only incline toward the cylinder axis slightly. However, provided the combustion system permits a weakly curved cylinder head bottom, this allows the cooling fins an even somewhat larger area above the valve bridge. Accommodating the maximum of cooling fins in the highly thermally stressed area of the valve bridge necessitates designing the gas ports' cross section to be relatively narrow and high. While four valve cylinder heads are virtually standard in air-cooled motorcycle gasoline engines, the extremely limited space above the cylinder head bottom precludes the implementation of a cylinder with four valves in air-cooled diesel engines.

Table 9-3 Cast aluminum alloys for air-cooled cylinder heads [9-44]

Alloy type	Alloy name	Alloy elements in % by weight								
		Cu	Ni	Si	Mg	Mn	Ti	Co	Zr	Sb
AlMgSiMn	Hydronalium, Ho 411, 511, Hy 418, 511, Hy 51, Hy 71	0 – 1.0	0 – 1.5	0.7 – 1.8	3.5 – 6.5	0.1 – 1.0	0.1 – 0.2	– – –	– – –	– – –
AlCuNiMg	Y alloy, A-U4NT	3.5 – 4.5	1.7 – 2.3	0.2 – 0.6	1.2 – 1.7	0.02 – 0.6	0.07 – 0.2	– – –	– – –	– – –
AlCuNiCoMnTiZrSb	RR 350, A-U5NZr	4.5 – 5.5	1.3 – 1.8	0 – 0.3	0 – 0.5	0.2 – 0.3	0 – 0.25	0.1 – 0.4	0.1 – 0.3	0.1 – 0.4

The distinctive feature of the cast aluminum alloys of air-cooled cylinder heads is their particularly high high temperature strength and resistance to cyclic temperature. Both of these material properties important for a cylinder head's dimensional stability are obtained by complex alloy compositions (see Table 9-3), controlled cooling in the permanent mold and special postheat treatment. The multi-alloyed material RR350 high temperature strength of 230 N/mm^2 at 200°C is the highest by far and even its high temperature strength of 200 N/mm^2 at 250°C is good. Aluminum alloys' greater thermal expansion and ductility make the *danger of valve bridge cracks* disproportionately greater in air-cooled cylinder heads than in water-cooled cylinder heads: Two cast-in steel plates that function as expansion joints can keep the region of the cylinder head base between the valves largely free of tensile stresses when the cylinder head cools down and thus prevent valve bridge cracks (Fig. 9-17). *Thermal shock tests* [9-41] in which the valve bridge is warmed to 300°C and cooled to 100°C in approximately 2 min intervals are an important aid for the development of air-cooled cylinder heads.

Fig. 9-17 Steel plates cast in valve bridges to prevent valve bridge cracks in the cylinder head of an air-cooled diesel engine (Deutz AG FL 513)

9.1.4.4 Engine-integrated Cooling Systems

Comparison of the External Cooling Systems of Liquid-cooled Engines

By virtue of the principle, cooling systems in air-cooled engines are *engine-integrated* since the component heat dissipates directly into the ambient air. The engine and cooling system constitute a unit. Cooling fans with cowling and integrated lubricating oil coolers as well as equipment or vehicle coolers, e.g. the hydraulic oil coolers of construction equipment or transmission oil coolers of vehicles, are add-on engine parts. Air-cooled engines require smaller quantities of cooling air since they utilize it better (greater temperature increase). However, the narrower flow cross sections cause relatively high air velocities in the finning and thus relatively high cooling air resistances (Fig. 9-18). The adaptation of cooling systems to these conditions results in varying sizes and designs of components in water-cooled engines. Thus, air-cooled engines' fans are only approximately half as large in diameter as fans of comparable liquid-cooled engines. They are however operated at speeds two to three times higher and constructed somewhat longer because of the guide vane required. The radiators are also substantially more compact, their end faces on the cooling air side being up to 60% smaller than conventional radiators. They are usually mounted on an engine without elastic intermediate elements. This causes high mechanical stresses and requires aluminum radiators with low inertial forces and higher strength and rigidity. Efficient intercooling based on the air/air principle has been applied to air-cooled engines from the start. Thus, the charge air can be cooled far below the temperature of a liquid-cooled engine's coolant. Depending on the type of intercooler arrangement (before the cooling fan or in the parallel flow to the other cooling air consumers), charge air temperatures

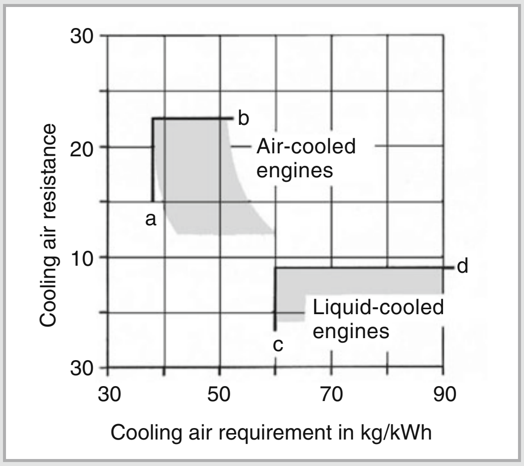

Fig. 9-18 Cooling systems of direct injection diesel engines: Operating ranges for cooling air resistance as a function of the cooling air requirement: **a** Maximum permissible cooling air heating, **b** Economically still justifiable cooling air resistance, **c** Maximum cooling air heating for conventional systems, **d** Maximum pressure increase of conventional fans

are reached, which are only 25–45 K above the respective ambient temperature.

Cooling Fan Designs and Design Criteria

An air-cooled engine's relatively high cooling air resistance and its requirement of a fan with the smallest possible dimensions and low speeds leads to axial cooling fans with aerodynamically highly loaded flow cascades. Basically accommodated in the flywheel, radial fans are reserved for small one and two-cylinder engines.

The axial fan's compact design facilitates simple cooling airflow routing. High efficiencies with low noise emission can be achieved when the flow cascade is designed and manufactured more meticulously. Two designs are distinguished based on the arrangement of the guide vane: Both bladings in fans with a guide vane downstream from the impeller (outlet guide vane fan) are deceleration cascades in which pressure builds up. When the guide vane is upstream (inlet guide vane fan), it is an acceleration cascade and lowers the static pressure so that the impeller alone must produce the pressure increase, compensating for the preceding pressure drop.

Given the multitude of constraints, a decision on the suitability of one of the two fan designs can only be made after weighing their respective features. Table 9-4 lists the most important features [9-42]. Both designs have proven themselves in the field. Overall efficiencies of 80–84% are achievable when certain values for the flow and pressure coefficient and the limits for a flow without separation in the axial cascade are observed [9-43]. The total power required for engine-integrated cooling is 2.5–4.5% of the rated engine power depending on the fan efficiency, cooling air requirement and resistance to flow.

Along with good aerodynamic design, the noise a fan generates is acquiring ever greater significance. It should not appreciably influence overall engine noise and must be free of tonal components. In light of mounting environmental regulations (see Sect. 16), this may often necessitate higher development costs. A fan's aerodynamic noise consists of three different components: The strongest noise source, turbulence and vortex noise extends over the audible frequency range. On the other hand, the tonal noise generated by the impeller with a multiple of the fundamental frequency of the human ear (number of blades times speed) is perceived as many times louder than an equally strong broadband noise. Uneven arrangement of the impeller blades provides a remedy.

The *sound power* emitted by the cooling fan can be described by the empirically grounded law

$$L = L_{sp} + 10\log[(\dot{V}/\dot{V}_0)(\Delta p_g/\Delta p_{g0})^2]$$

Table 9-4 Differences between the outlet guide vane and inlet guide vane fan

	Outlet guide vane fan	Inlet guide vane fan
Maximum efficiency	84%	80%
Minimum specific sound power level	31 dB(A)	33 dB(A)
Tool requirements for pressure die casting	Only two mold halves (bladings are free of overlap and allow axial demolding)	Two mold halves plus radial slide (bladings are not free of overlap)
Engine installation	High	Low
	Acoustic sensitivity during disturbances upstream from the fan (obstacle, flow constrictions)	

where the parameters of the operating point are $\dot{V}$ = volumetric flow and Δp_g = overall pressure increase and the reference variables are $\dot{V}_0 = 1\,m^3/s$ and $\Delta p_{g0} = 1\ mbar$. A constant characteristic for every fan design, L_{sp} is the specific sound power; the second term corresponds to the sound power of the operating point. Fan loudness can only be reduced in a defined operating point (cooling air requirement and the cooling system's resistance to flow) by decreasing the specific sound power. In addition to the flow coefficient and pressure coefficient, the magnitude of which may only vary in a certain range of values for optimal design, it is significantly influenced by the:

- type of guide vane arrangement,
- aerodynamic quality of the flow cascade and the fan inlet,
- position of the operating point to the design point,
- radial gap between the impeller and housing wall,
- axial distance between both bladings,
- type of impeller blade shape (radial or sickle),
- type of blade arrangement on the impeller circumference and
- number of blades and type of pairing of the number of blades.

The high quality axial fans of air-cooled engines now reach specific sound power levels of 31 dB(A) and are distinguished by low broadband noise.

A uniform temperature level of components and lubricating oil and thus optimal conditions for engine operation (consumption, exhaust gas quality, noise emission and service life) are obtained by controlling the quantity of cooling air. A control in which the cylinder head temperature is the controlled variable and, allowing for the lubricating oil temperature, remains constant is especially advantageous. A hydraulic clutch installed in the fan hub can control fan speed.

9.1.4.5 Examples of Implemented Diesel Engines

The range of commercially available air-cooled diesel engines extends from universally implementable small single cylinder direct injection diesel engines with their typical design preferred for construction equipment and power and pump units (see Sect. 18.1) up through powerful V12 diesel engines for heavy commercial vehicles (Fig. 9-19). The high performance variant with exhaust gas turbocharging and intercooling pictured here is part of a line of six, eight and ten cylinder engines and is tailored to the specific requirements of a 38 ton dump truck used at large construction sites and strip mines. It exemplarily demonstrates the advantages of an integrated engine cooling system, which not only encompasses the actual engine cooling but also the heat exchanger that cools the charge air, the engine oil and the oil utilized in the transmission and the retarder. Thus, the engine merely has to be connected to the fuel supply and the exhaust manifold when it is installed. In normal driving, the specific air requirement for engine cooling is 41 kg/kWh. The fan's power consumption requires

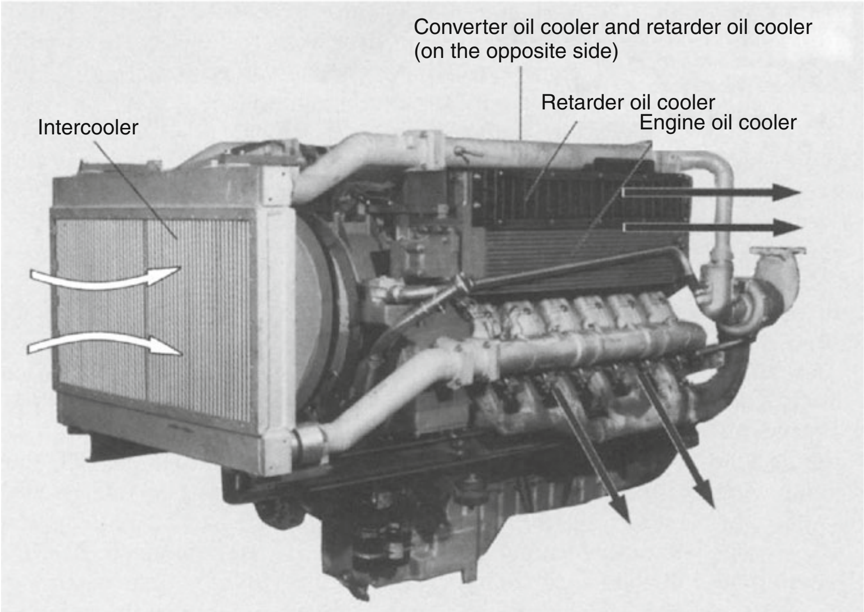

Fig. 9-19 Deutz AG BF12L523CP air-cooled twelve cylinder diesel engine, V_H=19.144 dm^3; P_e= 441 kW at 2,300 rpm

approximately 11.6 kW. Air-cooled industrial diesel engines are found in the mid and lower power range in between (see Sect. 18.2) and are employed for many purposes of installation including soundproofed diesel engines equipped with encapsulation (see Sect. 16.5).

9.1.4.6 Limits of Air-cooled Engines

The air-cooled diesel engines available in the power range up to 440 kW have reached a high level of maturity and are utilized preferably as industrial engines because of their simplicity and ruggedness (see Sect. 18.2). Air-cooled diesel engines remain little accepted in the motor vehicle sector because of their insufficient comfort, particularly their low supply of heating heat.

As supercharging rates increase, air cooling is being called into question as an equivalent, alternative cooling process more and more frequently since high mean effective pressures cause substantial mechanical loading of the cylinder unit and crankshaft assembly and a significant increase of the thermal loading of the piston, cylinder barrel and cylinder head. The limits of thermal loading ensue from the high temperature strength of the aluminum alloy for the cylinder head and the maximum dissipatable cooling energy. The disadvantage of smaller heat transfer coefficients h_K on the cooling side may be compensated by a larger heat gradient between a component and cooling air and by enlarging the exothermic component surface through thinner and closer fins on which there are also limits however. There are limits on a larger temperature gradient too since it inevitably increases component temperatures: Factoring in sufficient lubrication, the temperature ought to be a maximum of 190°C in the slide face of the liner, 240°C on average in the cylinder head bottom for dimensional stability and not above 280°C in the valve bridge.

Aluminum alloys' lower high temperature strength than gray cast iron's makes the cylinder head the weakest component and, thus, the component of an air-cooled engine that determines its performance. The cylinder head's dimensional stability determines the quality of the cylinder head seal. Given the cylinder barrel's increased temperature and the aluminum alloys' greater thermal expansion, thermally induced component deformations have proven to substantially exceed the mechanical deformations caused by forces and pressure [9-44]. In addition to a shift of the maximum sealing pressure on the inner sealing area resulting from the curved cylinder head bottom, the exhaust port's plastic deformation also observable in highly supercharged engines influences the sealing pressure distribution.

However, the higher temperature level of air-cooled engine components also causes greater warming of the aspirated combustion air and thus a lower cylinder charge. When their smoke emission is identical, air-cooled diesel engines' rated power in suction mode is approximately 2.5% lower than water-cooled engines' and 3.5% lower in the point of the maximum torque. A simulation determined that the hotter intake port causes approximately 50% of this, the higher wall temperatures of the cylinder liner cause approximately 30% and the hotter cylinder head and piston top each cause 10%. Since the temperature differences between the components and the combustion air is relatively small, no appreciable power loss occurs in supercharged engines without intercooling; it can be compensated by a marginally higher charge air pressure in engines with intercooling.

The higher temperature level of components that form the combustion chamber together with the higher temperature of the charge determines the final compression temperature and thus the maximum combustion temperature. As basic tests on cooling's influence on the formation of NO_X confirmed [9-45], this is decisive for the formation of NO_X, (see Sect. 15.3) and it complicates efforts to lower NO_X emissions as emission limits are continually tightened and causes considerable problems for air-cooled engines [9-46].

While the measures that

- intensify internal cooling (intercoolers and cooling gallery pistons),
- elevate component temperature limits necessitating the use of expensive materials, lubricants and more complex machining (cylinder head material with extremely high high temperature strength, fully formulated lubricating oils, cold worked cylinder barrel surfaces, Alfin bonded piston ring carriers and dual keystone rings with molybdenum coating) and
- insulate components against heat (thermal insulation of the exhaust port from the cylinder head)

could significantly shift the limits set by the restricted external heat dissipation in air-cooled diesel engines, they entail a higher level of technology that comparable water-cooled engines do not require at all or not to the same extent.

9.2 External Engine Cooling Systems

9.2.1 The Function of Engine Cooling Systems

9.2.1.1 Definitions

As explained in Sect. 9.1, the cooling system establishes an important prerequisite for trouble-free engine operation by dissipating heat from thermally critical points in engine components (cylinder head, piston, cylinder liner, etc.) and agents (engine oil, fuel, charge air, etc.) as a *cooling load* to the environment either directly (e.g. in air cooling, see Sect. 9.4) or, usually, by a closed coolant circuit and a radiator to comply with functional limit temperatures.

The cooling of agents is referred to as direct cooling when a heat exchanger releases the heat directly to the cooling air and as indirect cooling when the heat is released to a closed coolant circuit.

The arrangement of the components that dissipate heat to one another and their control constitute a cooling system.

From the perspective of thermal engineering, engine cooling for components consists of "heat exchangers" with usually small heat exchanging surfaces connected in parallel and/or a series. This, in turn, requires large heat transfer coefficients [9-47] (see Sects. 7.2 and 9.1).

Along with the radiator for the coolant, heat exchangers for

- engine oil,
- piston cooling oil or water,
- nozzle cooling with fuel, cooling water or cooling oil,
- charge air,
- exhaust gas recirculation and
- transmission oil

are also integrated in the coolant circuit.

Compromises often have to be made when designing a cooling system and its components. While cooling used to be primarily specified for rated and extra power, factoring in unfavorable boundary conditions (e.g. climate and seasons), the ever stricter requirements for fuel consumption, pollutant emission and idle and part load performance have made *control of the temperature of the components* and agents rather than their cooling the primary function of the cooling system. This not only necessitates providing sufficient cooling capacity but also using adjustable actuators such as thermostats, valves, pumps or fans.

9.2.1.2 Engine Cooling

Depending on the engine, the heat dissipated from engine cooling with the coolant as the cooling load is 40–100% of the rated power or 20–40% of the supplied fuel energy (see Table 9-5). High pressure supercharging's increase of the power density has led to a shift in heat balances as effective efficiencies have increased. When an engine's release of heat to the coolant decreases, the heat absorbed in the oil cooler and intercooler increases so that the overall cooling load remains approximately constant. On the other hand, the exhaust gas energy decreases as the effective brake work increases.

In turn, the cooled exhaust gas recirculation implemented to reduce NO_X introduces additional quantities of heat into the coolant as a function of the recirculation rates. This often increases the complexity of a cooling system considerably.

Regarded as energy loss, the release of the cooling load to the environment may be expediently utilized ecologically and economically by cogeneration to increase system efficiency (see Sect. 14.1).

9.2.1.3 Cooling Agents

The heat loss from transmissions and brakes in power train systems and vehicles must be dissipated too and the cooling system designed appropriately.

The additional cooling requirement relative to the input power to the transmission is

- 1–3% for mechanical transmissions
- up to approximately 5% for automatic transmissions with a torque converter
- 5% for hydraulic transmissions for locomotives when oil/water cooled (40% when air/oil cooled) and
- 25% for hydraulic transmissions for rail coaches (when the allowable temperature of the transmission oil is T = 80–100°C and only briefly 125°C to a maximum of 130°C).

Table 9-5 Dissipated heat fluxes in % of rated power

Engine type/speed range	Engine cooling[a]	Engine oil	Intercooling	Coolant heat	EGR cooling
Low speed engines 60. . .200 rpm	14. . .20	6. . .15.3[b]	20. . .35	40. . .70	–
Medium speed engines 400. . .1,000 rpm	12. . .25	10. . .15	20. . .40	40. . .80	–
High performance engines 1000. . .2,000 rpm	30. . .50	5. . .15	10. . .20	10. . .20	–
Commercial vehicle engines 1800. . .3,000 rpm with exhaust gas turbocharging and intercooling	30. . .50	30. . .50	15. . .30	45. . .80	10. . .20[c]
Naturally aspirated engines	50. . .70	50. . .70	n. a.	50. . .70	–

[a] Cylinder, cylinder head and exhaust gas turbocharging cooling.
[b] Lubricating oil and piston cooling oil cooling.
[c] Lubricating oil cooling (piston cooling by water).
Note: These and the following figures for quantities of heat, volumetric flows and temperatures are guide values. The diversity of engine design, power ranges and operating conditions explain the large range.

Table 9-6 Cooling water temperatures in °C

	Low speed two-stroke engines	Medium speed four-stroke engines	High speed four-stroke engines
Engine cooling water			
Inlet into engine	65–75	70–80 (82)	76–87
Outlet from engine	75–80	80–90	80–95 (110)
Temperature difference in the engine	5–10	5–10	4–8
Preheating to	50	40–50	40
Preheating during heavy fuel oil operation to	60–70	60–70	n. a.
Fresh water (sea water)			
Inlet into radiator, maximum[a]	32–38	32–38	32–38
Outlet from radiator, maximal	≤ 50	≤ 50	≤ 50

[a] Tropical operation.

Table 9-7 Volumetric coolant flows relative to engine power in l/kWh

	Low speed two-stroke engines	Medium speed four-stroke engines	High speed four-stroke engines	Commercial vehicle engines
Engine cooling water	6. . .15	30. . .40	50. . .80	50. . .90
Fresh water	30. . .40	30. . .50	30. . .50	N.A.

Hydrodynamic brakes in commercial vehicles (e.g. retarders) can introduce larger quantities of heat into the cooling system than the engine releases in normal operation.

The temperatures for the agents water, oil and charge air are a function of the engine's size, type and mode of operation. Engine cooling water temperatures are kept somewhat lower in large low and medium speed engines than in high speed engines. Making allowances for a bearing's construction, materials and design, lubricating oil temperatures are often significantly lower (Table 9-6). The dissipated heat fluxes and the desired or allowable temperature differences produce the volumetric coolant flows (see Table 9-7).

9.2.2 Cooling System Design

9.2.2.1 Cooling Systems with Direct Heat Dissipation

Of the two options to dissipate engine heat to the environment [9-48], direct cooling dissipates the cooling load to the environment in an open coolant circuit. This is part of

– air-cooled engines,
– engines with circulating water cooler,
– engines with cooling tower cooling and
– engines with evaporation cooling.

With a few exceptions, direct cooling is now synonymous with air cooling (see Sect. 9.1.4).

9.2.2.2 Cooling Systems with Indirect Heat Dissipation

Heat Dissipation

During indirect heat dissipation or cooling (Fig. 9-20), an engine initially releases its cooling energy to a coolant in a closed circuit (main circuit). A heat exchanger in the cooling system then transfers this heat to another coolant (secondary circuit).

Water-cooled engines are divided into:

– Engines with fan cooling (air/water cooling): These are used wherever cooling water is unavailable for the secondary circuit (autonomous cooling systems), i.e. primarily vehicle engines but also stationary engines. The advantage of zero water consumption must be obtained with the fan's consumption of power. High fan powers must be installed especially when the installation conditions in a vehicle are poor (< 1% of the rated power for cars, up to approximately 5% for medium commercial vehicle engines and approximately 10% for large commercial vehicle engines). So as not to unduly increase fuel consumption, such fans are regulated so that the maximum power consumption is only needed in critical cooling conditions. Air cooling systems do not have any components or lines that conduct

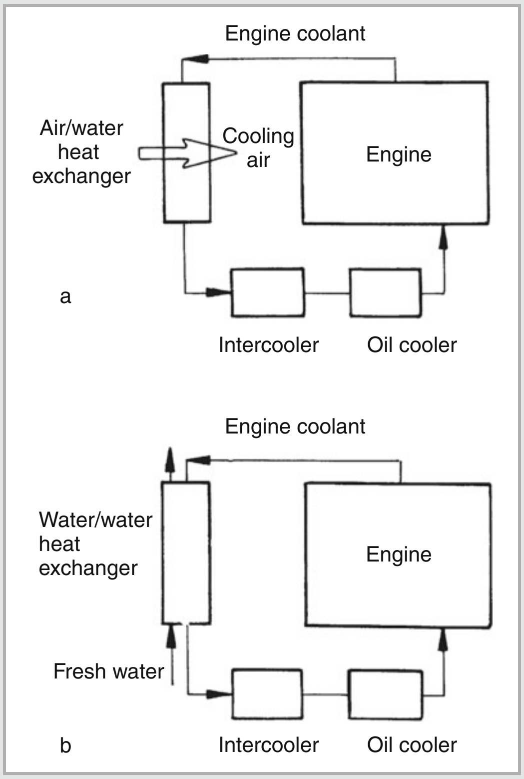

Fig. 9-20 Indirect cooling with a closed cooling system; **a** Air cooling **b** Water cooling

fresh or sea water. However, the higher system costs and larger space requirements are drawbacks.

– Engines with water/water cooling in a closed secondary circuit: The secondary circuit's fresh water (external water) is conducted from above into a cooling tower where it is distributed over a large area or atomized in a natural or fan generated counterflow air draft and releases its heat to the air by evaporation and cooling. The cooling capacity depends on the air temperature, flow rate and humidity. The water loss is approximately 3%. The greater complexity of equipment, problematic antifreeze protection and plume formation are disadvantages. This type of cooling only makes sense for large plants.
– Engines with water/water cooling in an open secondary circuit: The fresh or raw water circuit (secondary circuit) is open. The water is freshwater (river or lake water) or sea water or brackish water. Fresh open water, especially sea water, constitutes a virtually infinite heat sink. However, this cooling has a number of problems in practice, which must be taken into account when designing cooling circuits. Hull-mounted and keel cooling are special forms of this type of cooling (Fig. 9-21).

Cooling System Design

Main and secondary circuit cooling: The heat exchangers are connected in parallel on the secondary side, i.e. the external water side (Fig. 9-22).

Single, double and multiple circuit cooling: The heat exchangers in these systems are connected in a series on the secondary side. Several configurations are possible:

– *Single circuit system:* In a single circuit system, the components being cooled to the desired temperatures (cooling

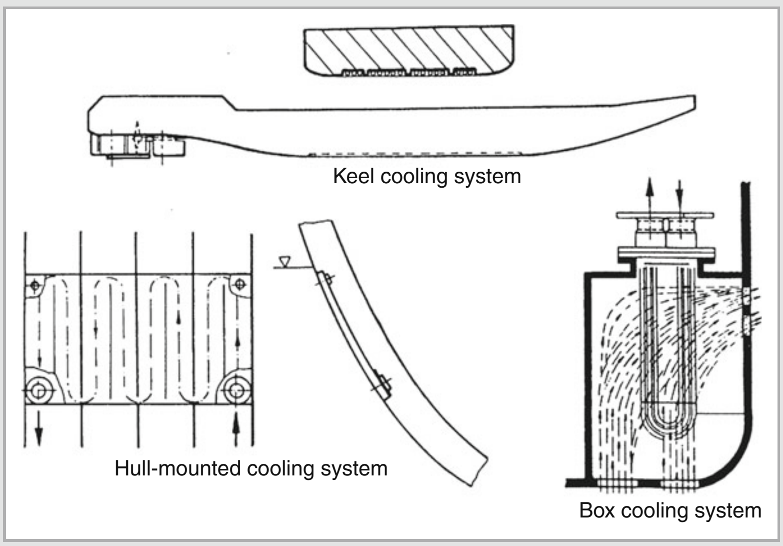

Fig. 9-21
Water cooling for inland ships

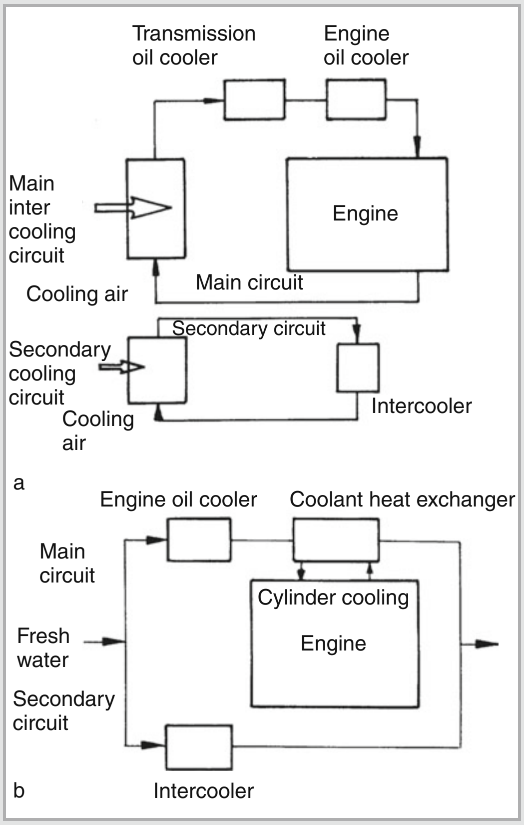

Fig. 9-22 Main and secondary circuit cooling: The cooling system is connected in parallel on the coolant side (secondary side). The greater part of the heat is dissipated in the main circuit, the lesser part in the secondary circuit: **a** Cooling with air; **b** Cooling with fresh water

priority) are connected in a series and/or parallel in the cooling circuit (Fig. 9-23). The heat exchangers in the cooling circuit interact. When configured appropriately, this is exploited for the purpose of "self-regulation" [9-48, 9-49].

– *Double circuit system:* A better adaptation of the cooling system to the variables such as engine load, heat requirement or fresh water temperature is obtained by thermally separating the circuits into a high and low temperature circuit (HTC and LTC). LTC and HTC heat exchangers are connected in a series on the fresh water side (on the cooling air side in vehicle engines) (Fig. 9-24). This also has the advantage of functioning with smaller volumetric fresh water flows. The allocation of the heat flux to the subcircuits depends on the design of the system: Which heat exchangers are in which branches [9-48]? Thus, for example, vehicle engines and marine and generator engines often have a mixed double circuit system in which the air intercooler is placed upstream from the air/water heat exchanger.
– *Mixed double circuit system:* In this cooling system, also known as an "integrated cooling system", the HTC and LTC are hydraulically connected (Fig. 9-25). The HT cooling water is cooled by mixing it with the LT cooling water. This eliminates the HTC heat exchanger. However, this requires meticulous design work to ensure that the desired coolant flow is present in every branch of the circuit. In compact engines with fully integrated accessories, one branch of the HT and one of the LT circuit are united (Fig. 9-26). Approximately two thirds of the coolant flow passes through the high temperature circuit (HTC) and the rest through the low temperature circuit (LTC). The HTC cooling water is cooled by mixing it with the LTC cooling water and not by a heat exchanger. At low and part load, the LTC is controlled so that the heated engine cooling water reaches the intercooler directly through a bypass line. The subcircuit remains uncooled and heats the charge air. The engine cooling water is increasingly sent through the heat exchanger and the intercooler as the engine temperature rises [9-50].

Combining these features and placing the heat exchangers in the individual branches of the coolant circuits furnishes many options to design and optimize a cooling system based on the particular engine design and the placement of the cooler in the secondary circuit. Sea water operation in particular must allow for contamination and deposits [9-51].

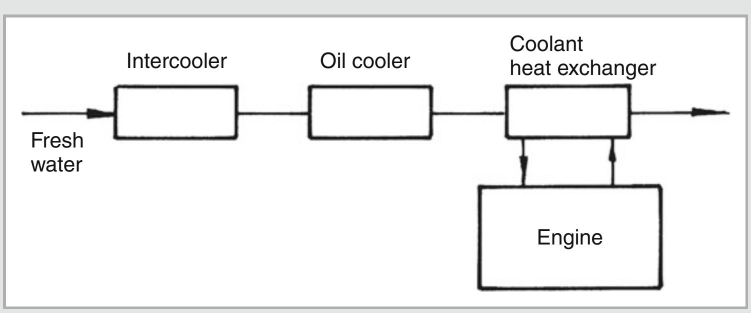

Fig. 9-23
Single circuit cooling

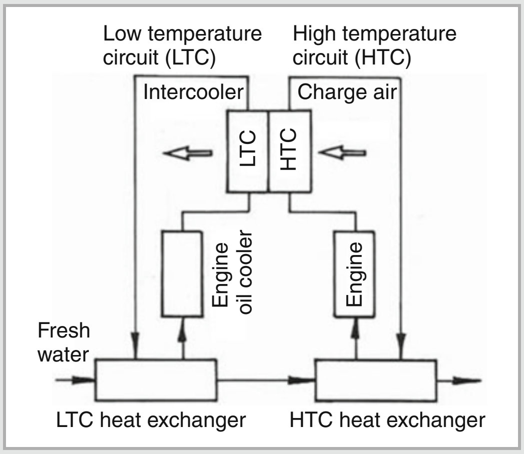

Fig. 9-24 Double circuit cooling with a high (HTC) and low temperature circuit (LTC) and with fresh water cooling

Cooling systems for sea water operation: These can be divided into:

– "Conventional" sea water cooling (Fig. 9-27): The technically simple thermal engineering advantage of greater inlet temperature differences and lower pump outputs resulting from smaller volumetric sea water flows entails accepting the disadvantages of sea water operation.
– Central cooling (Fig. 9-28): In light of the disadvantages of conventional cooling, central cooling in which a large central cooler cools the fresh water of all other heat exchangers is usually preferred. The high technical complexity generated by more as well as larger coolers (lower inlet temperature differences and additional heat transfer resistances) and more lines and pumps is also an indicator of the problems of indirect sea water cooling. On the other hand, fewer lines and components are needed to conduct sea water. Since the temperature in the fresh water circuits remains largely constant, the cooling system proves easier to control. Therefore, central cooling is advantageous, above all, for multiple engine systems.

Intercooler placement: The placement of the intercooler in the cooling circuit is determined by the demand for a minimum charge air temperature at full load and by the need for an acceptable charge air temperature at the particular engine operating points. There are several solutions to choose from:

– *Internal intercooling*: Intercoolers integrated in the engine cooling water circuit (main circuit) on the coolant side produce a certain self-regulation of the charge air temperature (Fig. 9-29). While this can only cool the charge air to the level of the engine cooling water temperature at best (power losses), the engine load adjusts (improved engine running properties). The charge air is cooled in the upper load range and heated in the lower.
– *External intercooling*: The intercooler/s is/are placed in a secondary circuit and coolant flows through it/them parallel to the other heat exchangers (Fig. 9-23). Thus, a large inlet temperature difference is on hand, which can cool down charge air even more. Efforts have also been being made for roughly twenty years to exploit this advantage for smaller engines (commercial vehicle engines). Since the charge air is thermally decoupled from the engine, appropriate control of the external circuit should be utilized to adjust the temperature to the engine's load level.
– *Two and multistage intercooling*: The intercooler is divided into one subcooler connected to the high temperature circuit and one subcooler connected to the low temperature circuit (two-stage intercooling in Fig. 9-24). When the

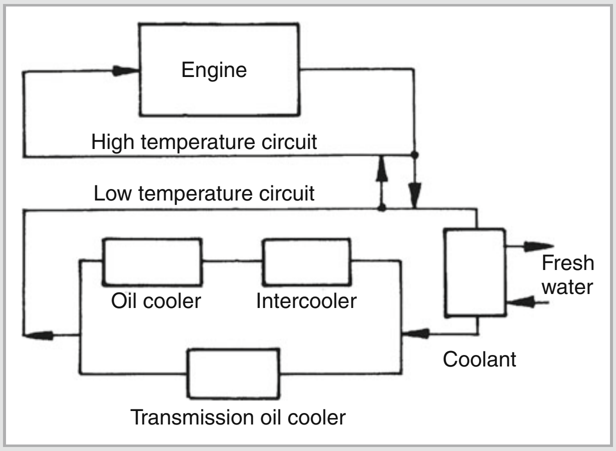

Fig. 9-25
Mixed double circuit cooling: High and low temperature circuit are hydraulically connected

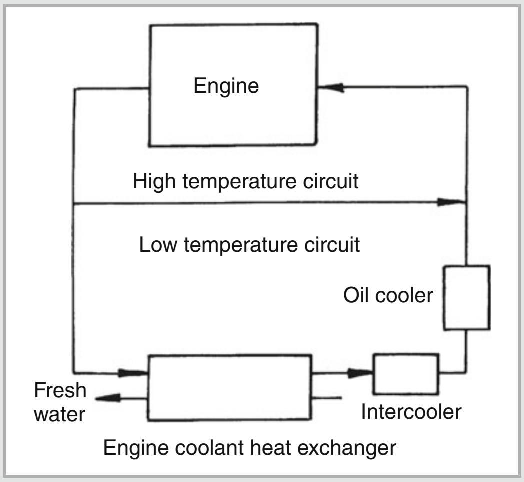

Fig. 9-26 Mixed double circuit cooling with combined branches of the high and low temperature circuit

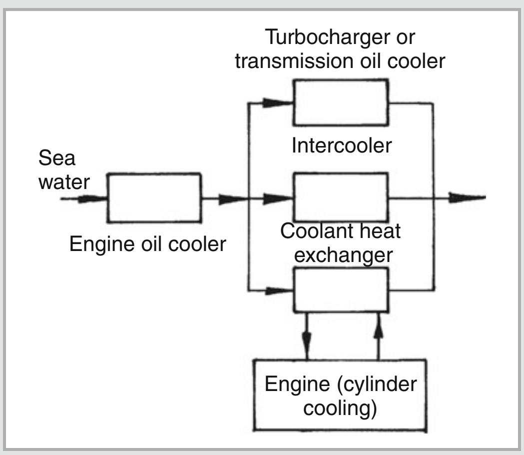

Fig. 9-27 "Conventional" sea water cooling (single circuit system)

charge air enters the cooler in the HT circuit, its high temperature can increase the coolant temperature for heat consumers (e.g. fresh water producers) to 98°C and correspondingly increase the useful heat gradient. The majority of the charge air heat is dissipated in the HT circuit (1.5–2.6:1). The low temperature stage is switched off in part load operation ($\leq$ 40%, particularly at full speed) or in arctic operation and the charge air is heated by the engine cooling water in the HT circuit. This translates into better combustion, lower firing pressures and less smoke emission. Engine cooling water heating may be switched in when the engine load drops below 15% [9-48].

Such complexly configured cooling systems are implemented with numerous flow branches and unions and components with different flow resistances (see Sect. 9.2.4). The pressure and volumetric flows are modeled for individual branches of the circuit and verified in a test of the system or when it is installed. Where necessary, the individual partial flows may be corrected with adjustable restrictors [9-52].

Cooling Systems in Vehicle and Compact Engines

Engines for vehicles, mobile or transportable generator units and all types of fast boats assume a special position. Since compact design is required, the cooling system components also have to be integrated in the engine assembly or situated as closely to the engine as possible [9-53, 9-54].

The actual engine cooling water cycle in vehicle engines or engines with similarly compact designs includes the engine itself, radiator, lines, expansion tank, temperature controller and circulation pump. Powered directly by the engine, the engine cooling water pump forces the cooling water through the heat exchangers (oil cooler and intercooler), which are each connected in a series or in parallel, into the engine's cooling chambers (cylinders and cylinder heads) and through other components being cooled, e.g. exhaust gas turbochargers, exhaust manifolds, etc., or through other heat exchangers, e.g. oil coolers, exhaust gas recirculation coolers, etc.. The cooling water flow is divided according to the cooling capacity requirement. A temperature controller ensures that the cooling water in a cold engine reaches the suction line to the engine cooling water pump completely or partially through a line that bypasses the radiator (Fig. 9-29). This ensures that the operating temperature is reached quickly and the desired temperature maintained even when engine load varies.

Positioned at the highest point of the cooling circuit, the cooling water expansion tank collects the changes in cooling water volumes when temperatures fluctuate and removes exhaust gas (e.g. during cylinder head gasket blow-by) and air (e.g. inlet through water pump gasket) from the cooling circuit. Air bleeding and venting of the cooling circuit is so important because air or gas in the cooling water reduces throughput. The cycle becomes unstable at an air content of 12% and collapses at 15% [9-55]. Moreover, heat transfer is diminished or a corrosion attack facilitated. In addition, the expansion tank contains a coolant reserve for smaller leaks and creates a certain pressure buffer.

A combined pressure and suction valve provides the pressure equalization necessary for excess pressure produced when the cooling water is heated and when a vacuum is produced when it is cooled. Continuous bleed lines run from the engine – as well as from the air/water radiator in vehicle engines – to the expansion tank. In addition, a connecting line runs from the expansion tank to the suction side of the cooling water pump.

The cooling water's flow through the cylinder and cylinder head is laid out meticulously. In engines with a water chamber shared by all cylinders, defined *flow conditions* must ensure

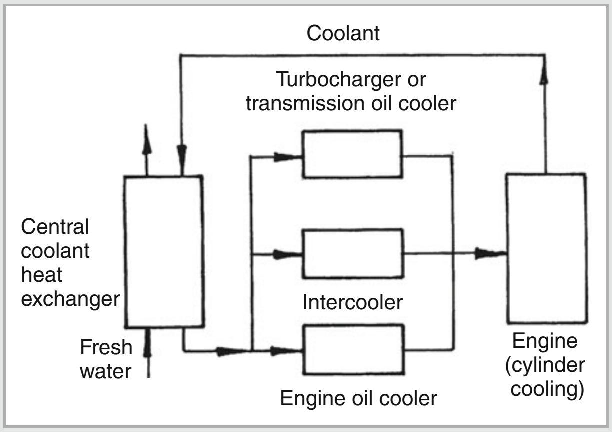

Fig. 9-28
Central cooling

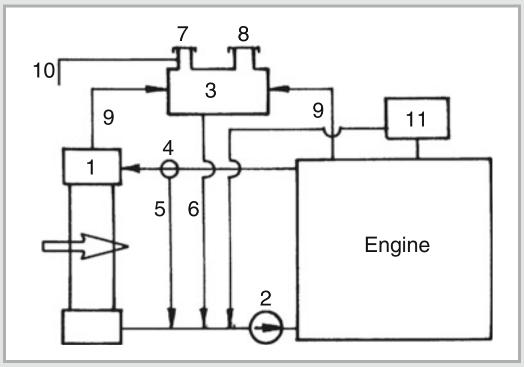

Fig. 9-29 Commercial vehicle engine cooling. *1* Radiator; *2* Water pump; *3* Expansion tank; *4* Thermostat; *5* Bypass line; *6* Filling line; *7* Working valve; *8* Fill cap with safety valve; *9* Bleed lines; *10* Overflow line; *11* Heat exchanger for cab heating

that the cooling water flows uniformly around the thermally loaded zones (cross flow). On the one hand, the flow velocity must be high enough for heat exchange. On the other hand, cavitation should not occur. Pressure losses should remain low as well. Conditions prove to be simpler in large single cylinder engines because the cylinders are individually supplied from a manifold. Since cylinder heads have a complicated structure in which aspects of thermal and mechanical strength and castability are prominent, experience is required to design the cooling water paths in them so that temperatures that are as uniform as possible set in. Neither leakage nor the venting of water when they are heating up should cause air pockets to form. Local boiling should not cause any aggregation of vapor bubbles. The cooling water flow is optimized with flow simulations (CFD) and can be tested in flow tests on a Plexiglas model.

9.2.3 Cooling System Control

9.2.3.1 Control Requirements

With an eye toward fuel consumption, wear, pollutant emission and noise, efforts are being made to reach the engine operating temperature quickly, to maintain desired temperatures regardless of engine load (preventing undercooling and overheating) and to lose a minimum of power through additional assemblies (e.g. cooling fan) in order to sustain cooling. Moreover, the coolant circuit must be able to adjust to the demands of engine operation and additional boundary conditions. In addition, the coolant circuit must not only adapt to the cooling load, which changes as the engine load changes, but also different air temperatures (e.g. seasonal and geographic conditions).

Engines are also utilized to different capacities, e.g. full load, part load and idle. In an initial approximation, the heat that accumulates from the engine cooling water, intercooler or engine oil cooler is a function of the engine power. The heat exchangers' cooling capacity is primarily determined by the coolant's mass flows in the secondary circuit and the fluid being cooled. Thus, the cooling capacity requirement and supply diverge whenever the coolant flow changes disproportionately to the engine power.

Since the heat exchangers and the entire cooling system are designed corresponding to an engine's rated power for the maximum cooling capacity required (cooling load), it must be assured that the desired (intended) temperatures are obtained even when the engine operating points diverge. Thus, control of coolant temperatures is indispensible. The temperature of the coolant is regulated in the main circuit.

Return temperature: The coolant temperature in the engine outlet serves as an *index for the thermal state of the engine*. The actuating element is a thermostatic control valve equipped with a temperature sensor. The desired temperature is set by mixing cooled cooling water conducted past the

radiator in the bypass with uncooled cooling water in the control valve. The process is similar for oil coolers.

Flow temperature: The coolant temperature is regulated at the inlet. Implemented similarly, mixed temperature control is primarily applied to multiple engine systems with a central cooling system. Larger engines have electrically or pneumatically actuated PI controllers; smaller engines function with controllers without auxiliary power (expansion element). The engine cooling water's thermal inertia may be disregarded when the engine load changes; the cooling water reaches the operating temperature after 50–70 seconds, even in large medium speed engines. When their region of travel changes, marine engines use water pumps with pole reversing electric motorss to adapt the flow rate of sea water to the actual cooling requirements.

Preheating: Engines from which full power is demanded immediately after starting (especially quick-starting and no-break standby generating sets) must be preheated or kept heated, to be precise, by heating the engine cooling water to 40°C by means of electric preheating equipment or thermal heat exchangers. Larger and large engines are usually preheated (see Table 9-6).

9.2.3.2 Fan-Cooler Combinations

Fan-cooler combinations, i.e. primarily for vehicle engines and stationary units, use the fan speed to control the flow of cooling air to establish the desired cooling water temperature. This can effectively reduce energy consumption and noise emission. The temperature set by the fan is higher than the thermostat's operating range. The fan may be powered:

- *Mechanically:* The rigid connection of fan speed to engine speed does not allow any control intervention and is thus disadvantageous. When bimetal viscous clutches are employed, the speed can be regulated as a function of the cooling air temperature (based on the vehicle radiator). Modern electrically controlled clutches can control fan speed as a function of any reference variable.
- *Electrically:* Fan speed is independent of engine speed. The fan can continue running when the engine is stopped. This provides flexibility in the arrangement of the fan and radiator.
- *Hydraulically:* A hydraulic variable speed coupling produces the drive. The control range is limited. The fan must be placed directly at the engine. All in all, this is a complex solution primarily used for larger vehicle engines when the power input for the fan can no longer be transmitted by a belt without problems and vibrations in the drive train need to be damped.
- *Hydrostatically:* Higher power inputs can also be transmitted over greater distances. Thus, the radiator can be positioned flexibly. The control range and the power loss are large, unit volume and mass on the other hand are small.

There are various options for the design of mechanically powered fans. Axial fans with small gaps (approx. 8 mm) are used for maximum demands on efficiency and air mass flow. Given the engine motion, such a small gap necessitates firmly installing the frame ring around the fan on the engine. Were the frame not firmly on the engine, the relative motion would have to be compensated by a large gap (20–30 mm). Shrouded fans are used to prevent a strong reverse flow through the gap and flow dispersion. In addition to higher fan gap tolerances, this arrangement has advantages for acoustics and flow stability [9-56].

9.2.4 Implemented Cooling Systems

9.2.4.1 Cooling Systems for Commercial Vehicle Diesel Engines

Figure 9-29 illustrates the standard cooling system described here. The heat exchanger output (approximately 6–10 kW) needed to heat the cab is not incorporated into the design of the radiator circuit. However, to prevent the engine cooling water temperature from dropping too greatly, no more than 20% of the heat may be drawn from the cooling circuit. The heat from the engine cooling water in highly boosted engines may be insufficient to cover the heat requirement, particularly in raised roof cabs (approx. 20 kW). In such cases, an additional heating unit must be designed in. Other heat exchangers (for retarders and transmission oil) are connected in line with the engine between the water outlet out of the engine and the thermostat.

Extreme requirements are imposed on the radiators of armored vehicles, which must dissipate sizeable heat fluxes from the engine and transmission under extreme ambient temperatures (−30°C to +45°C). This requires a large cooling air mass flow and a large gradient from the cooling water temperature to the outside temperature, e.g. 95°C (110°C) at an outside temperature of 20°C (45°C). The cramped space in a tank requires an extremely compact radiator design. In addition, the fan must overcome high intake and pressure losses at correspondingly higher fan output (e.g. 13.5% of the engine power of 1,100 kW in a Leopard II tank).

9.2.4.2 Diesel Locomotive Cooling Systems

The cooling system of the Deutsche Bahn's 215, 218 and 210 series diesel locomotives is constructed as a two circuit system with a high and low temperature circuit (HTC and LTC) (Fig. 9-30). The actual cooling system consists of two radiator cores configured in a V-shape through which a hydrostatically or hydrodynamically powered axial fan aspirates the cooling air. The fan speed is regulated so that the coolant temperature remains nearly constant as a function of the engine load and outside temperature. A mixing valve (with a response temperature of 30°C) hydraulically connects the LTC and HTC to reduce undercooling of the LTC when outside temperatures are low. A second mixing valve (approximately 60°C) enables preheating both cooling circuits and keeping them warm by a heater and a circulation pump. The quantity of

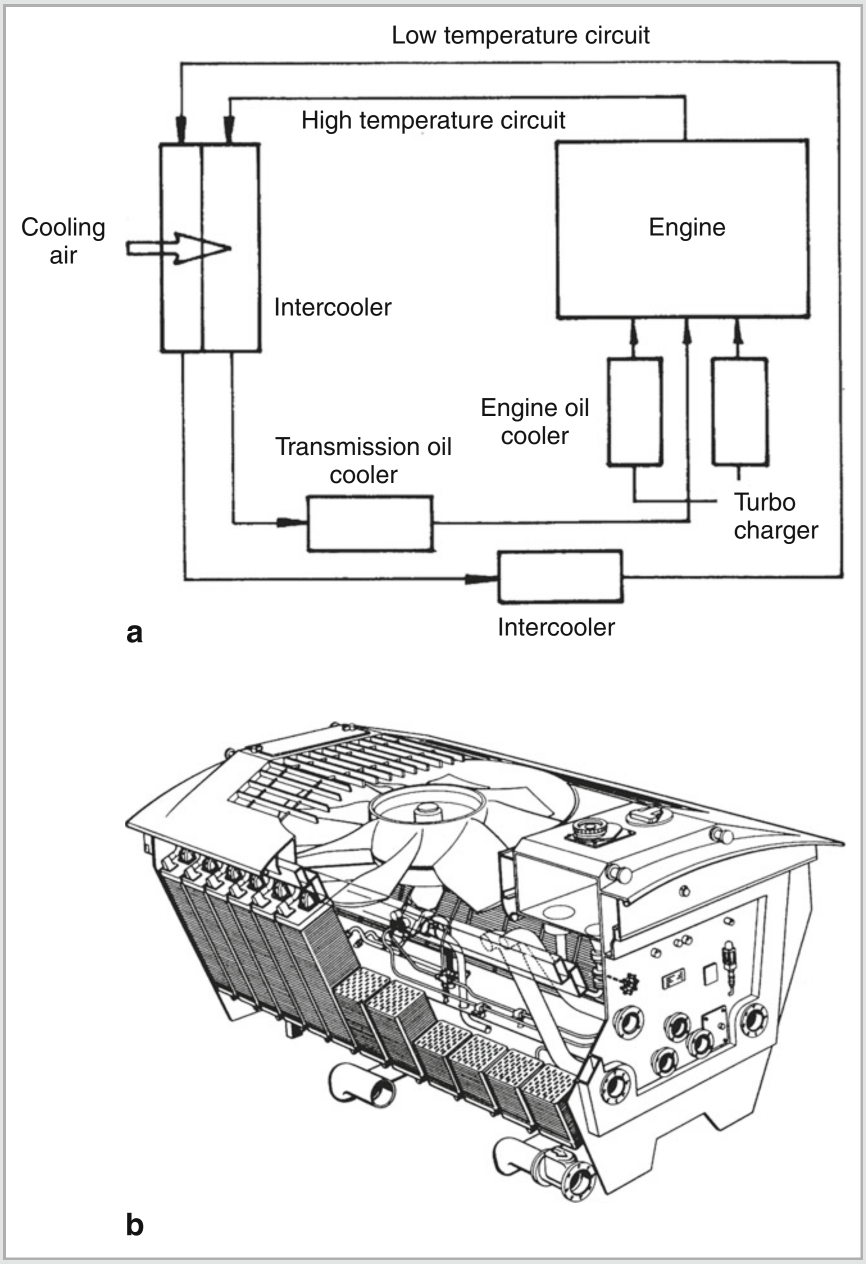

Fig. 9-30
Radiator of Deutsche Bahn's model 215, 218 and 210 diesel locomotives (double circuit cooling with LTC and HTC; see Fig. 9-24): **a** schematic of the cooling circuit; **b** radiator in V configuration

coolant that flows from each one of the circuits to the other is returned to the expansion tank [9-57, 9-58].

9.2.4.3 Cooling Systems for Marine Propulsion Systems

Figure 9-31 is a detailed schematic of a marine engine system's central cooling system with a hydraulically coupled HTC and LTC (integrated cooling system) including pumps, expansion tanks and other accessories. The principle configuration corresponds to Fig. 9-26. This cooling system is the basis for the MAN B&W 40/54 to 58/64 engine series.

9.2.4.4 Vehicle Cooling System Cooling Modules

In vehicles, the basic components necessary for cooling are united in a cooling module that normally consists of a radiator, intercooler, air conditioner condenser and fan

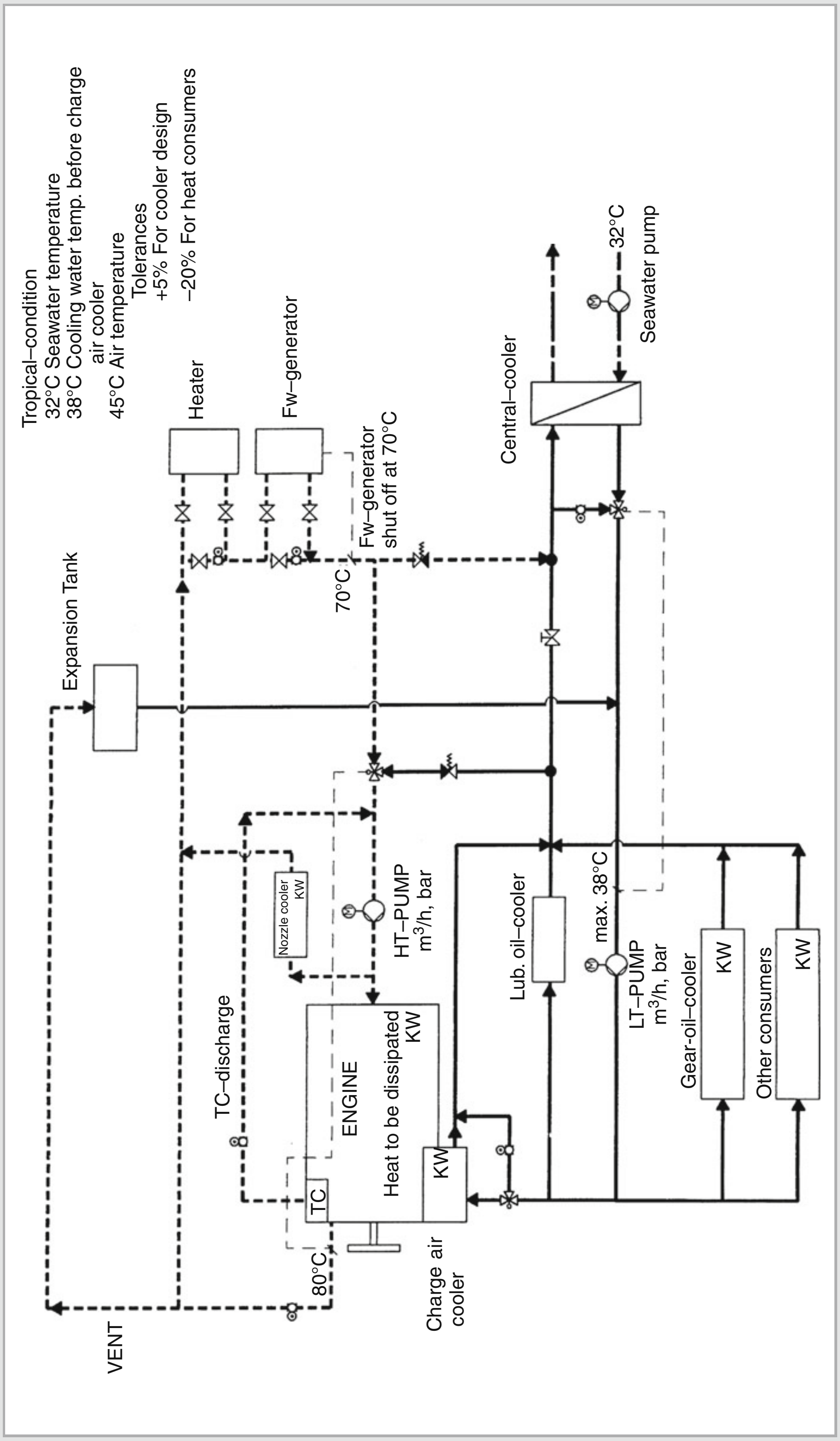

Fig. 9-31 Integrated double circuit central cooling system of a marine propulsion system (MAN B&W)

cowl. The fan may be powered electrically or mechanically directly by the engine. Hydrostatic fan drives are frequently used in busses and special vehicles.

The arrangement of the heat exchangers in the cooling air flow ensues from the particular inlet and target temperatures of the fluids being cooled. Normally, this results in the sequence of condenser, intercooler, radiator and fan.

It is important that air guides and sealings prevent heated air from being recirculated in order to optimally utilize the cooling module's installed cooling capacity.

9.2.5 Heat Exchangers

9.2.5.1 Thermal Engineering

Only stationarily operated heat exchangers with fixed baffles between two fluids, so-called *recuperators*, are treated here [9-59].

The index "1" always designates the exothermic fluid and the index "2" the heat absorbing fluid. The index "e" stands for "inlet" and the index "a" for "outlet".

Thus, the following applies to the cooling of the mass flow $\dot{m}_1$:

$$\Delta T_1 = T_{1e} - T_{1a} \geq 0 \tag{9-1}$$

and the following to the heating of the mass flow $\dot{m}_2$:

$$\Delta T_2 = T_{2a} - T_{2e} \geq 0 \tag{9-2}$$

Inserting the heat capacity flows

$$\dot{W}_1 = \dot{m}_1 c_{p1} \tag{9-3}$$

$$\dot{W}_2 = \dot{m}_2 c_{p2} \tag{9-4}$$

yields the following for the heat transfer capacity

$$\dot{Q} = \dot{W}_1 \Delta T_1 = \dot{W}_2 \Delta T_2 \tag{9-5}$$

When the power is related to the largest possible temperature difference in the heat transfer, i.e. the difference of the two mass flows' inlet temperatures

$$\Delta T_e = T_{1e} - T_{2e}, \tag{9-6}$$

then, with $\dot{W}_{\min}$ as the smaller of the two heat capacity flows, the following is obtained

$$\frac{\dot{Q}}{\Delta T_e} = \frac{\dot{W}_1 \Delta T_1}{\Delta T_e} = \frac{\dot{W}_2 \Delta T_2}{\Delta T_e} = \dot{W}_{\min}\varepsilon \tag{9-7}$$

with

$$\varepsilon = \frac{\Delta T_1}{\Delta T_e} \quad \text{for } \dot{W}_{\min} = \dot{W}_1 \tag{9-8}$$

or

$$\varepsilon = \frac{\Delta T_2}{\Delta T_e} \quad \text{for } \dot{W}_{\min} = \dot{W}_2 \tag{9-9}$$

In this relationship, ε is denoted as the *exchanger efficiency* or *operating characteristic* [9-60, 9-61].

It can be demonstrated that ε is a function of the number of heat transfer units

$$N = \frac{kA}{\dot{W}_{\min}} \tag{9-10}$$

the ratio of the heat capacity flows

$$R = \frac{\dot{W}_{\min}}{\dot{W}_{\max}} \tag{9-11}$$

and the flow configuration in the heat exchanger [9-61, 9-62]. In Eq. (9-10), kA is the product of the coefficient of heat transmission k and the area of heat exchange A, which cannot be separated expediently in finned surfaces – as usually exist here.

The VDI Heat Atlas [9-63] contains formulas and diagrams for different flow configurations (Fig. 9-32). It and [9-62] and [9-64] specify a mathematical method, the cell method, which may be used to calculate exchanger efficiency for any flow configurations.

The task when designing heat exchangers is to calculate the necessary exchanger efficiency ε from Eq. (9-7) at heat capacity flows specified on the basis of Eqs. (9-3) and (9-4) and the relative heat transfer capacity $\dot{Q}_c/\Delta T_c$ to be obtained. Depending on the intended flow configuration, the number of heat transfer units N and thus the kA value required according to Eq. (9-10) can be determined from the related formula or the relevant diagram. In many cases, N cannot be calculated explicitly from the specified formulas. Thus, it must be determined iteratively.

Since the magnitude of the heat capacity flows is often dependent on the selected size and design of the heat exchanger, e.g. a vehicle radiator dynamically pressurized with cooling air, an assumed size of the heat exchanger is postulated in practice and iterated by the mean fluid temperature until a satisfactory correspondence has been obtained. When necessary, the thusly calculated heat exchanger output serves as the basis to adjust the size of the heat exchanger to the performance requirements.

The heat transfer coefficients h_1 and h_2 and the coefficients of thermal conductivity of the baffle λ_w and the fins λ_{r1} and λ_{r2} are needed to actually calculate the kA value. Then, the Péclet equation applies:

$$\frac{1}{kA} = \frac{1}{h_1 A_1} + \frac{\delta_W}{A_W \lambda_W} + \frac{1}{h_2 A_2} \tag{9-12}$$

A_w being the area of the flat baffle, δ_w its thickness and A_1 and A_2 the heat exchanging surfaces, rated in part with a fin efficiency η_r [9-65]. The heat transfer coefficients are normally specified as Nusselt numbers

$$Nu = h \cdot l/\lambda = Nu(\text{Re}, \text{Pr}) \tag{9-13}$$

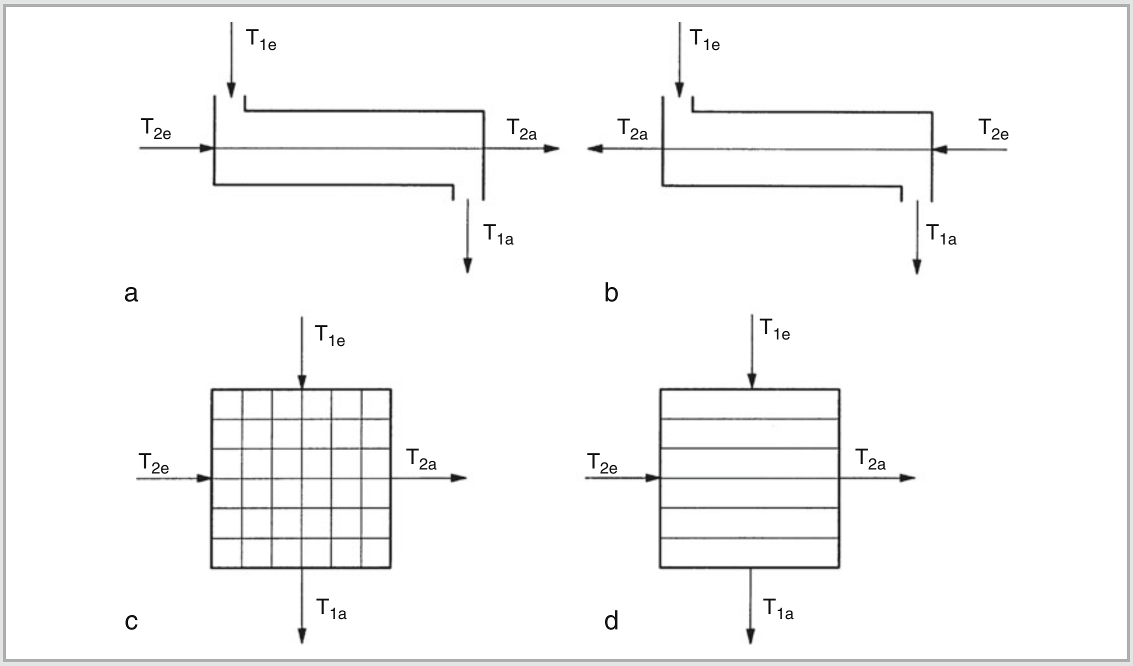

Fig. 9-32 Some basic forms of flow configuration in heat exchangers: **a** pure parallel flow; **b** pure counter flow; **c** pure cross flow; **d** cross flow cross-mixed on one side

The following

$$\mathrm{Re} = \upsilon \cdot l / \nu \tag{9-14}$$

is the Reynolds number and

$$\mathrm{Pr} = \rho \cdot \nu \cdot c_p / \lambda \tag{9-15}$$

is the Prandtl number, l being a characteristic dimension of the fixed body, λ the coefficient of thermal conductivity of the flowing medium, υ its flow velocity, ν its kinematic viscosity, ρ its density and c_p its isobaric mass-specific heat capacity.

Heat exchangers for motor vehicle engine cooling are characterized by very compact designs. The transfer area A_1 or A_2 relative to the unit volume V of the heat exchanger matrix is typical. Values for this are generally available:

$$\frac{A}{V} > 700 \frac{m^2}{m^3} \tag{9-16}$$

However, the kA values relative to the matrix volume V are even more informative. Table 9-8 specifies some recommended values.

9.2.5.2 Radiators

A distinction is made between air-cooled and liquid-cooled radiators. Air-cooled heat exchangers used for vehicle diesel engines should be designed very compactly and lightweight but the drops in pressure allowable on the air and coolant side limit their compactness. Moreover, the finned tube core's sensitivity to contamination increases as the surface density on the air side increases in accordance with Eq. (9-16) and thus also establishes limits, especially for tractors and construction vehicles for instance. Finned tube cores consist of combinations of fins and tubes, which are differently shaped depending on the manufacturing process. A brazed radiator design that consists of flat tubes and corrugated fins brazed together is pictured in Fig. 9-33.

Mechanically joined radiators (Fig. 9-34) have round or oval tubes stuck through precisely dimensioned holes in louvered fins. These are flared mechanically to produce lasting contact pressure between the tube and hole to conduct heat from the tube to the fin. In both cases, the fins are provided with a multitude of gill-like notches to improve the heat transfer [9-66]. Made of glass fiber reinforced polyamide, the dispensing and receiving tank is interconnected with the particular header by an elastic seal. Solutions for brazed radiators with brazed-on aluminum receiving tanks also exist. The tubes may be provided with turbulence inserts to prevent laminar flow that might occur when coolant throughputs are small.

Mechanically joined radiators cost less to manufacture than brazed radiators. However, their heat transfer capacity relative to the air side end face or the air side drop in pressure

Table 9-8 Common maximum mass flows for specific cross sections and attendant kA/V values for a high performance heat exchanger. Fluid 1 exothermic, Fluid 2 heat absorbing

Fluid 1	Fluid 2	Specific mass flows		
		$\dot{m}^*_1$ kg/(m^2 · s)	$\dot{m}^*_2$ kg/(m^2 · s)	(kA/V) · 10^{-3} W/(m^3 · K)
Coolant[a]	Air	250	12	370
Lubricating oil[b]	Air	150	6	200
Lubricating oil[b]	Coolant[a]	150	200	1,300
Charge air	Coolant[c]	12	40	280
Charge air	Air	20	10	100

[a] Mixture of water and glycol with a volumetric content of 70/30% at approximately 100 °C.
[b] Essolub HDX 30 at approximately 130 °C.
[c] Same as [a)] but at approximately 70 °C.

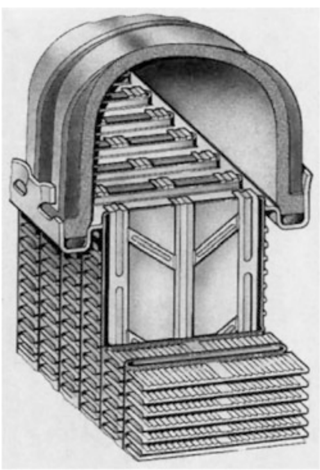

Fig. 9-33 Design of an aluminum brazed radiator for use in motor vehicles (source: Behr)

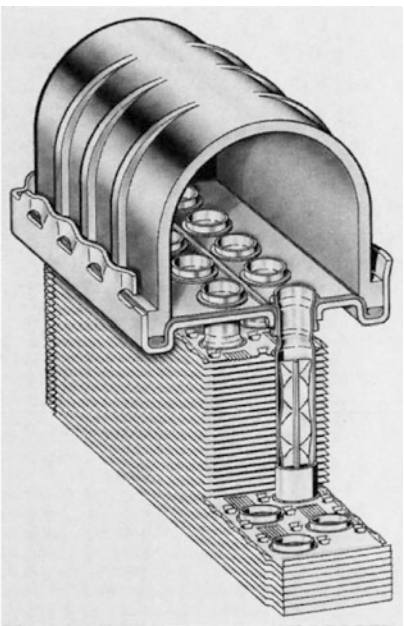

Fig. 9-34 Design of an aluminum mechanically joined radiator for use in motor vehicles (source: Behr)

is normally poorer. This makes larger radiators or higher fan powers necessary [9-67, 9-68].

Motor vehicle applications in European cars and commercial vehicles now almost exclusively use aluminum radiators.

Another air-cooled radiator design is the radiator core section pictured in Fig. 9-35 for a cooling system with a main and secondary circuit (see Sect. 9.2.2). A partial flow coming from the main circuit is cooled further in the secondary circuit in order to use it obtain a low charge air outlet temperature in an intercooler for instance. The core consists of louvered, usually smooth copper fins with holes for the flat copper tubes. A layer of lead-tin solder on the tubes connects tubes and fins. The headers, the receiving and dispensing tank and the cast connectors brazed to it consist of brass. The tube-header is connected and the tank is brazed on with soft solder. Very rugged and insusceptible to contamination, such radiator elements are preferred in rail vehicles but also employed in stationary engines. Figure 9-36 illustrates the use of simple turbulent flow radiator core sections in an underfloor cooling system for the Deutsche Bahn.

Particularly vibrostable designs have to be selected for radiators mounted directly on engines. Deutz AG utilizes an air-cooled radiator in a shell or plate design for its new

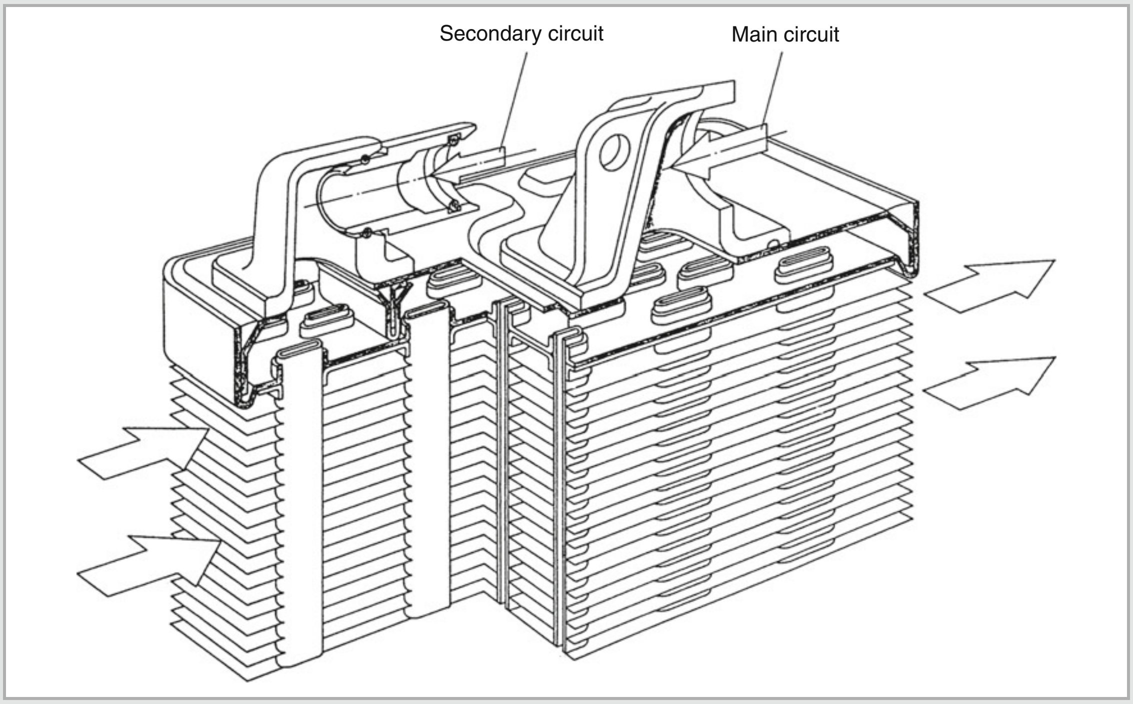

Fig. 9-35 Design of a nonferrous metal radiator core section with a main and secondary circuit in rail vehicles (source: Behr)

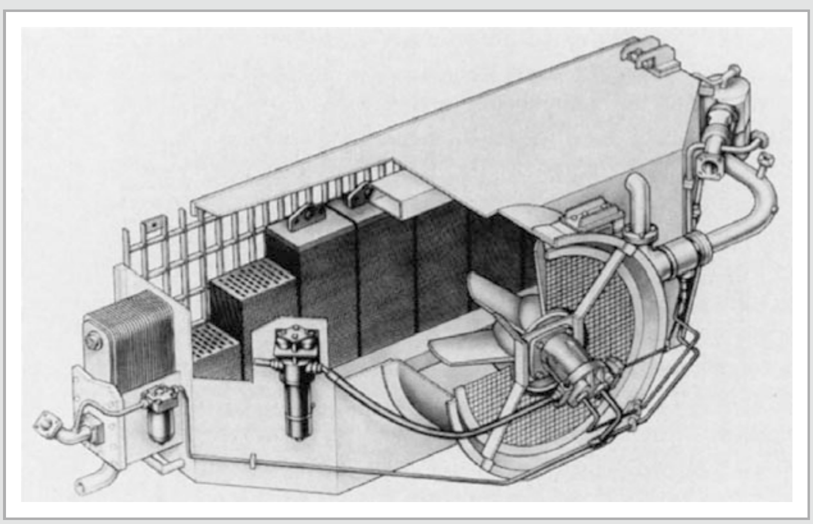

Fig. 9-36
Underfloor cooling system for a Deutsche Bahn railcar, consisting of ten radiator core sections with lengths of 700 mm and widths of 200 mm and a liquid-cooled transmission oil cooler in plate design and a hydrostatically driven fan. 405 kW are transmitted to the cooling air flow at an inlet temperature difference of 58 K (source: Behr)

generation of engines with an engine-integrated cooling system (see Sect. 18.2). Each of the elements that conduct coolant are formed of two aluminum half plates. Aluminum corrugated fins similar to those pictured in Fig. 9-33 are arranged between the individual elements. Outfitted with appropriate end plates and connectors, the entire radiator is brazed. Performance can be easily modified by the number of elements.

Liquid filled radiators are primarily employed in marine engines but also in part in stationary engines. Unfinned systems are normally employed since the heat transfer coefficients are similarly high on both sides of a heat exchanger. What is more, low sensitivity to contamination and good options for cleaning and resistance to corrosion are important when sea water is used. In addition to the classic tube bundle units [9-59], plate heat exchangers made of CuNi or titanium are used. They are usually installed separately from the engine but may also be integrated in the engine as in MTU's newer 595 engine series [9-69].

Figure 9-37 presents another sea water filled radiator design, which is also integrated in the engine. A flat tubular cooling unit is incorporated in a cast aluminum housing through which engine coolant flows. The sea water flows around the flat tubes in a register configuration and comes into contact with the CuNi components that are brazed together. The individual flat tubes are stamped with dimples, which are brazed to the dimples on the opposite side of the tube. This produces the mechanical strength required for the operating pressure of 6 bar gauge pressure as well as better heat transfer than in smooth tubes.

Fig. 9-37 Sea water filled radiator in flat tubular design for mounting on an engine (source: Behr)

9.2.5.3 Oil Coolers

Oil coolers must be designed for higher operating pressures than radiators, i.e. between 10 and 20 bar. In addition, their heat transfer from the oil to the wall surfaces is far lower. This always requires having to work with turbulence generating structures on the oil side. Soldering them to the wall surfaces also achieves the requisite increase of internal mechanical strength.

Air-cooled oil coolers are made of aluminum virtually without exception. Figure 9-38 pictures an oil cooler in the flat tubular design used for motor vehicle diesel engines. The design of the finned tube core resembles that of brazed radiators. However, the tanks are made of aluminum because of the increased operating pressures. As Fig. 9-39 shows, the turbulence plates are formed as offset fins.

Instead of flat tubes, aluminum plates between which the turbulence plates are arranged are also partially in use. Brazed-in as a boundary, plates with a rectangular profile produce channels that conduct air and oil. Such pack construction has the advantage of making it possible to manufacture any radiator designed without special tools. The type-specific tanks are welded onto the complete brazed heat exchanger block.

Another variant of an air-cooled oil cooler is the plate design. Here too, turbulence plates brazed with the plates on the oil side are indispensible for reasons of strength and heat exchange.

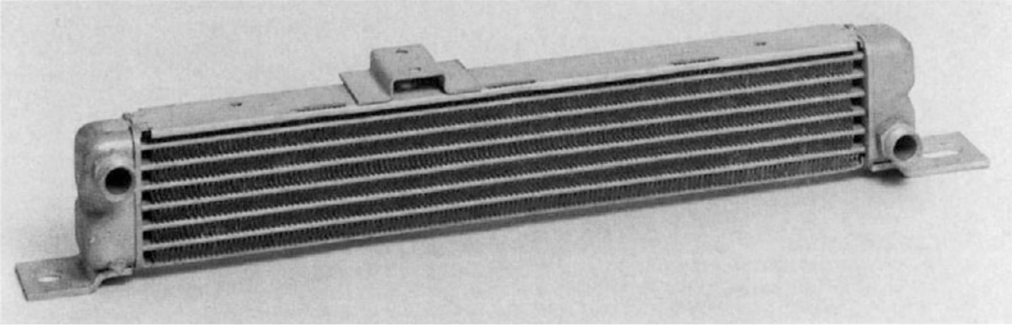

Fig. 9-38 Air-cooled oil cooler in flat aluminum design for use in motor vehicles (source: Behr)

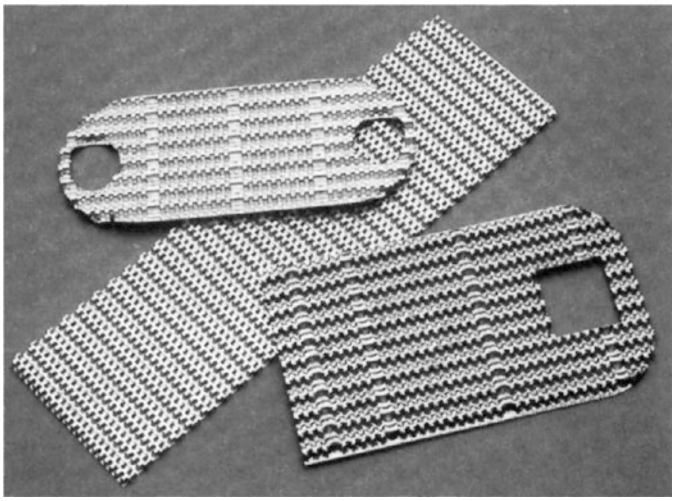

Fig. 9-39
Different turbulence plates for use in air and liquid-cooled oil coolers. Constructed of aluminum or steel (Behr)

Pictured in Fig. 9-40, aluminum stacked plate oil coolers are now only used for liquid filled oil coolers in car diesel engines. Thanks to the elevated edges of the plates, each of which is brazed with the next plate, these radiators do not need a separate housing. Turbulence inserts similar to the plates pictured in Fig. 9-39 are located on the coolant side and on the oil side or the plates already have a distinct structure that assumes this function. Such radiators are frequently bolted to the filter module with an oil filter as well.

The plate design illustrated on the right in Fig. 9-41 is frequently employed for heavy duty engines, e.g. for commercial vehicle applications. It has a longitudinal plate oil cooling unit in a special housing as pictured in Fig. 9-36 or incorporated directly in the engine block. The oil flows parallel through the individual plates equipped with turbulence plates and the coolant flows around the plate packs in the counter or cross flow.

The cut open round stack plate oil cooler pictured on the left in Fig. 9-41 can be installed under the oil filter and thus

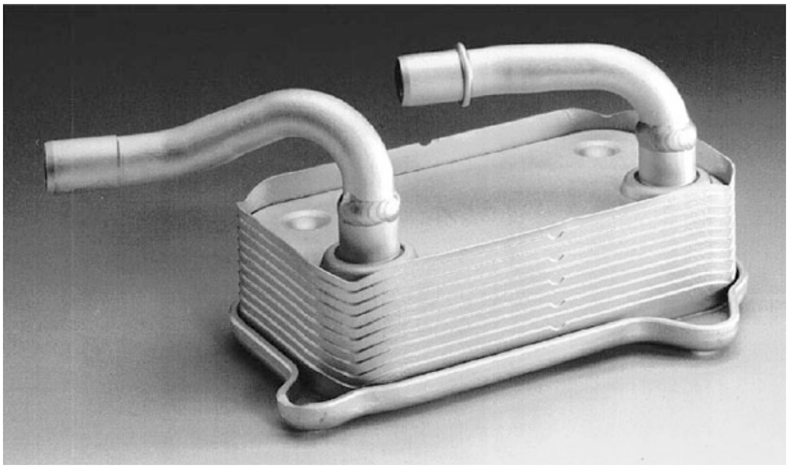

Fig. 9-40
Stacked plate oil cooler for car applications (source: Behr)

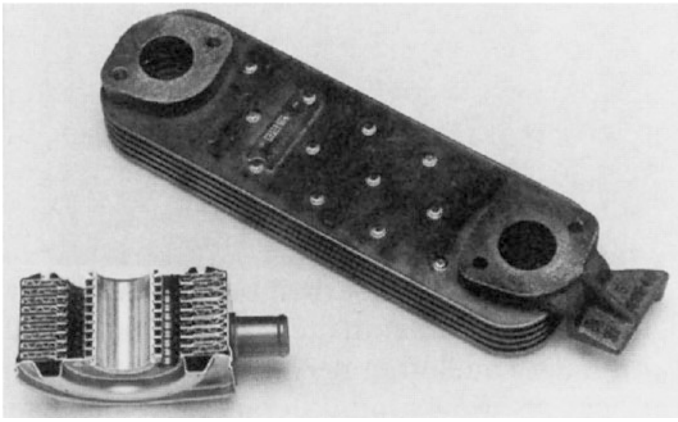

Fig. 9-41
Liquid-cooled oil cooler in plate design for installation under the oil filter (split, left) and in the engine block or a separate housing (right) (source: Behr)

integrated in the oil flow. Since the use of steel as a material entails higher weight, this product is now being used more and more rarely. Plate oil coolers are also occasionally manufactured entirely of aluminum or a CuNi alloy when sea water is used.

Not only plate oil coolers but also tube bundle heat exchangers are often implemented in marine and larger industrial engines. Allowing for oil's poorer heat exchange properties, the tube bundles are provided with intensive louvered finning over which oil flows based on the arrangement of the baffles. Another design utilizes smooth tube bundles and conducts the oil through tubes provided with special packing. Engine-integrated plate heat exchangers are also utilized for oil cooling [9-69]. However, since they are only furnished with a primary area of heat exchange, such units only have a limited capability to counteract the imbalanced heat exchange properties of oil and engine coolant or sea water.

9.2.5.4 Intercoolers

Intercooling increases the air density by reducing the air temperature. Thus, a larger mass of air is introduced into the combustion process. This results in higher engine power. Hence, the pressure drop Δp_1 on the charge air side has to be kept as low as possible when an intercooler is being developed. Otherwise, the pressure drop can overcompensate for the increase in air density induced by cooling and the intercooler causes a reduction in engine power. An efficiency factor

$$\eta_\rho = \frac{\dfrac{T_{1e}}{T_{1a}}\left(1 - \dfrac{\Delta p_1}{p_{1e}}\right) - 1}{\dfrac{T_{1e}}{T_{2e}} - 1} \tag{9-17}$$

is introduced to evaluate the density recovery obtained in the intercooler [9-70]. The temperatures T_{1e}, T_{1a}, T_{2e} and T_{2a} have to be inserted in *K*. p_{1e} is the absolute pressure of the charge air at the inlet into the radiator. η_ρ=1 in an ideal, infinitely large heat exchanger without any drop in pressure. However, η_ρ can also assume negative values, i.e. the density decreases, when high pressure drops occur at low boost pressures.

For the most part, air-cooled intercoolers are implemented in motor vehicles. They are placed upstream from the radiator and, in part, in other positions in passenger cars. The construction of an aluminum radiator core resembles a brazed radiator. As in radiators, glass fiber reinforced polyamide tanks are used when charge air temperatures are below 190°C. Particularly durable plastics can even be used at up to approximately 220°C. At even higher temperatures, permanent mold cast aluminum tanks must be welded on as pictured in Fig. 9-39. The flat tubes contain inner finning, which, making allowance for the pressure drop, are only intended to moderately increase the surface area.

Unless they are used in other vehicle or stationary engines, air-cooled intercoolers are normally constructed as an aluminum pack as described for oil coolers in Sect. 9.2.5.3. The fins on the charge and cooling air side have a design similar to that in Fig. 9-42. In turn, the advantage of pack construction is the ability to cost effectively manufacture small quantities of compact coolers with individual dimensions. Flat tubular systems made of steel or nonferrous metal are also used wherever space and weight play a subordinate role. To facilitate a low pressure drop on the charge air side, the flat tubes often do not contain any inner finning. The cooling capacity required must then be achieved by designing the radiator end face large enough.

Liquid-cooled intercoolers are preferable for charge airflows that facilitate flow and thus prevent pressure losses. They can be harmoniously integrated in an engine's charge air line [9-69, 9-71]. In addition, they are built even more

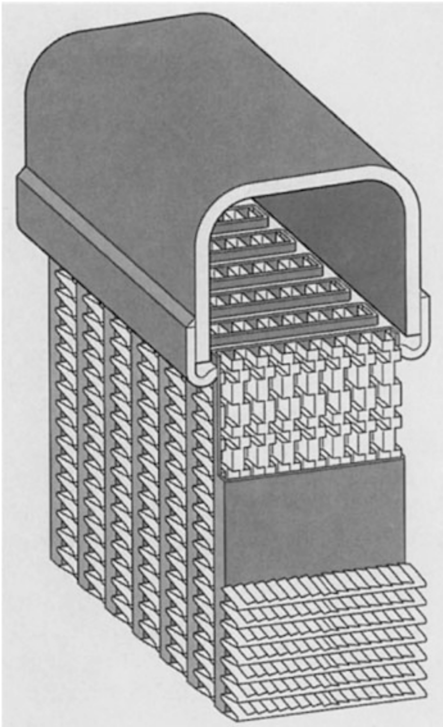

Fig. 9-42 Design of an aluminum brazed intercooler for use in motor vehicles (source: Behr)

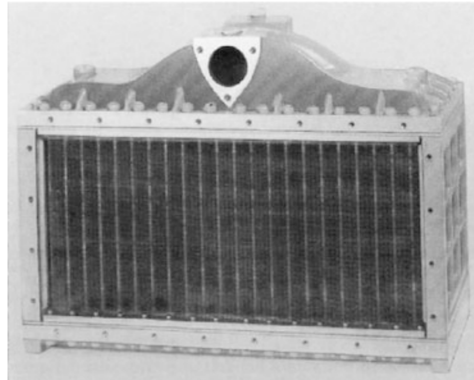

Fig. 9-43 Sea water filled, round tube intercooler for installation in the charge air line of marine diesel engines (Source: Behr)

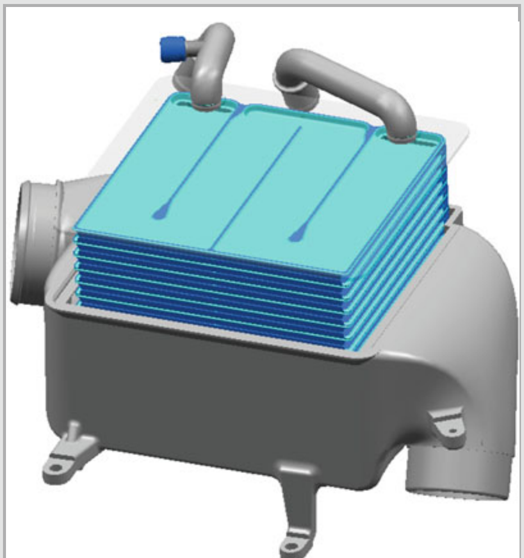

Fig. 9-44 Intercooler/radiator for the Mercedes S Class and Maybach (12 cylinder bi-turbo, M275)

compactly than air-cooled designs because their kA/V values have to be larger (see Table 9-8) and the ratio R of the heat capacity flows is better. Moreover, a cross-counter flow configuration is often implemented to obtain high exchanger efficiencies.

With the exception of motor vehicle engines, larger boosted diesel engines and marine engines in particular are now the main field of application for liquid-cooled intercoolers. This technology is also already being increasingly implemented in motor vehicles. Figure 9-43 pictures such a cooler with a heat exchanger matrix that intercools a marine diesel engine. It consists of round tubes with an outer diameter of 8 mm and lightly corrugated, louvered copper fins with a thickness of 0.12 mm. The fins and tubes are connected by flaring the tubes as in mechanically joined radiators (see Fig. 9-34). A CrNi or CuNi alloy is employed for the tubes depending on the individual case. The steel or special brass tube plates are connected by rolling in the tubes. Gray cast iron or an AlSi or CuAl alloy is selected for the receiving and dispensing tanks depending on the coolant used. Since the charge air and coolant temperatures can differ considerably, one of the plates is designed as a sliding plate, partly to prevent thermal stresses. Aluminum can then be utilized for the cast side sections. Otherwise, gray cast iron is preferred to obtain ratios of expansion similar to those in the tubes.

A liquid-cooled intercooler Mercedes Benz has been using for several years in its twelve cylinder biturbo engine that powers the Mercedes S Class and the Maybach is illustrated in Fig. 9-44. This intercooler is made completely of aluminum. The heat exchanger matrix consists of plates, each plate pair

forming a narrow channel for the coolant routing. Fins that absorb the heat of the charge air and release it to the coolant are arranged between these plate pairs. The housing is made of cast aluminum.

Other design variants exist for commercial vehicles.

Generally, liquid filled intercoolers are also suitable as *charge air preheaters* and are used as such with the identical design.

9.2.5.5 Exhaust Gas Heat Exchangers

Designs, Forms of Construction

Exhaust gas heat exchangers are gas/water (oil) heat exchangers. So far, every form has been designed with varying success, i.e. smooth and finned tubular heat exchangers with the gas flow configured in the tubes (gas tube design) and the external gas flow oriented vertically and horizontally. The diversity of designs arises from the cramped installation conditions (vertical and horizontal designs or the use of finned tubes from air cooler production), from manufacturing advantages when designs are derived from production air/water coolers or from particular price constraints. On the other hand, there are clear rules for engineering exhaust gas heat exchangers for diesel engines, which stem from regulations [9-72] and various tests. Specialist companies automatically observe these rules:

– Basically only straight tubes with inner exhaust routing are used, thus enabling mechanical cleaning and simplified repair.
– Tubes are attached by welding them in the tube plates. Rolling is inadequate when local overheating occurs and weakens rolled fits.
– Possibilities for the gas tubes to thermally expand are provided. For example, the tubes are given freedom to deflect from straight lines and directions of deflection are specified by small bends or an adjustable tube plate or flexible housing is designed in.
– Designs are primarily vertical, i.e. as far as possible, outlet chambers under the tubes with the exhaust flow directed downward are employed so that agglomerated particulates can separate out more easily and, where necessary, accumulating condensation can discharge. Since greater heights for maneuvering are required above the heat exchanger, disadvantages for mechanical cleaning may have to be accepted.
– Tube diameters not smaller than 12 mm Ø are selected.
– The design compels the cooling water to flow reliably at the exhaust gas inlet tube plate and in the inlet region of the gas tubes.
– The cooling water flow is always designed from bottom to top and good air bleeding is provided even in operation.
– Using circular flanges, exhaust inlet and outlet chambers are designed so both sides open easily or sufficient openings are designed in so that soot or potentially present solid metal sulfate deposits can be removed and the condensate outlet unblocked.
– Since exhaust gas valves never close completely, a low water flow is always incorporated to absorb heat from leaking exhaust gas.
– Given the reserve of wall strength usually present in commercial semi-finished tubes, small gas tubes (12–30 mm Ø) are constructed of high-grade steel X10CrNi MoTi 18.10, material no. 1.4571, and large tubes even of mild steel, St 37.10. Tube plates are usually produced from mild steel for reasons of cost and manufacturing. High-grade steel X10CrNiMoTi 18.10, material no. 1.4571, is also selected for exhaust inlet and outlet chambers. The materials 1.4539 and 1.4404 are also used for exhaust tubes in vehicle applications.

Figure 9-45 presents a vertical exhaust gas heat exchanger manufactured in small lots [9-73]. Horizontal models for smaller powers are also constructed for special installation conditions, low machine shops or spaces without package volume above the exhaust gas heat exchanger. Since there are still constraints on the recovery of diesel engine exhaust gas and the quantities are usually small, special exhaust gas heat exchangers are generally manufactured individually for each system in job production. Exhaust steam generators on the other hand are special exhaust gas heat exchanger designs that are also designed horizontally and vertically. They must be constructed for the individual case by special firms and equipped with considerably expanded safeguards.

Problems with Hot Side Fouling

The exhaust gas heat exchangers (exhaust gas/water heat exchangers) employed in diesel engines often register clogging of their gas passages with dry and loose to slightly moist deposits of soot particulates and liquefiable components. Therefore, common gas/water heat exchanger designs have only been effectively employed with additional measures.

Since soot formation in diesel engines is system induced, not even large engines with partly very low soot emission values are fully soot-free despite all the advances in combustion technology (not even after a soot filter!).

Research studies [9-74–9-76] have largely been able to explain the deposition mechanism in systematic experiments: Accordingly, the particles entrained in exhaust gas are subject to different forces (Fig. 9-46). Thermophoresis, i.e. the attraction of particles moving in the direction of the temperature gradient, and the adhesion of condensing hydrocarbons, water and sulfuric or sulfurous acids to the wall (adsorption) cause particles to accrete.

Exhaust pulsations and turbulence intensity in the exhaust gas cause the particles to be transported away again with the exhaust flow. In addition, drying of the wall layer holds promise as "cleaning".

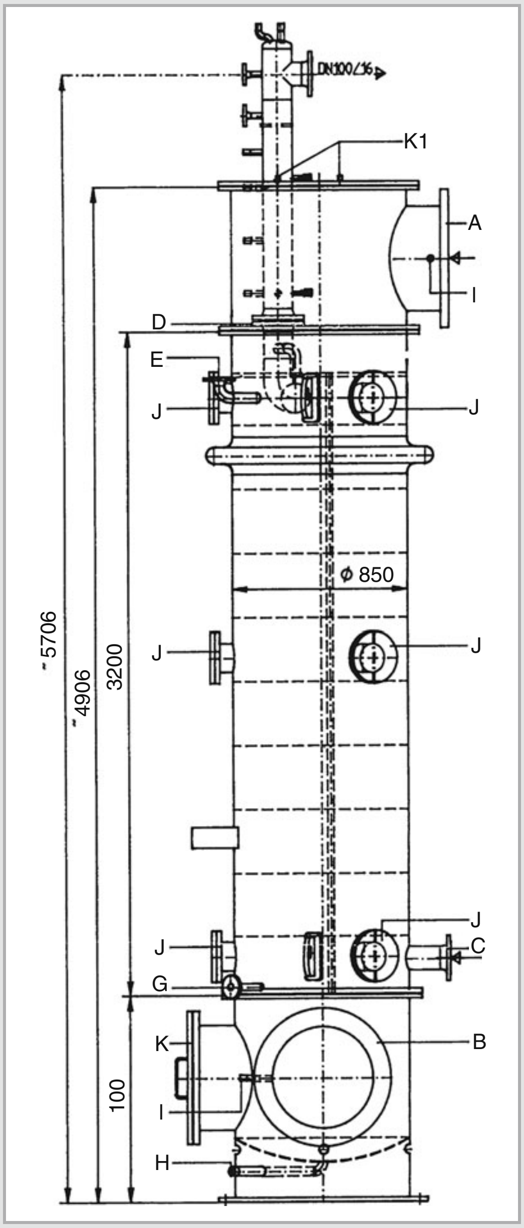

Fig. 9-45 Vertical exhaust gas heat exchanger (MBN, Neustadt/Wied). Heat output 718 kW to heat water from 100 °C to 120 °C at TA1=475 °C; TA2=180 °C; $\dot{m}_A/\dot{m}_W$ = 8475/3087 kg/h/kg/h; **A** Exhaust inlet and **B** Exhaust outlet; **C** Water inlet and **D** Water outlet; **E** Safety valve; **J** Inspection holes; **K, L** Water spray nozzles

While findings acquired in systematic experiments reveal potentials for improvements, not all difficulties have been eliminated. Exhaust heat recovery from diesel engines still entails the following problems:

– Without outside influence, fouling can gradually cause clogging.
– Fouling extremely impedes or prevents heat exchange from the exhaust gas to the heat recovery medium, e.g. water.
– Rather than leaving the heat exchanger steadily, particles agglomerate to larger particulates that are expelled aperiodically and pollute the environment.

Not only soot particulates but also metal sulfates (especially $Fe_2(SO_4)_3 \cdot H_2O$), which deposit in exhaust gas heat exchangers as voluminous solids in temperature ranges of the dew point of sulfuric acid/water, may contaminate the heating surfaces. These metal sulfates are formed by gradually detaching – especially from iron – from the engine's exhaust system for instance. The more sulfur is contained in the fuel and, in particular, the more SO_3 is contained in the exhaust, the more intensely they appear. This process increasingly manifested itself as oxidation catalytic converters were used in diesel engines in which increased SO_3 forms from SO_2.

Solutions

Both research studies and experience with the manufacturing of diesel engine plants deliver principle measures to prevent or decrease soot deposits. Moreover, periodic mechanical or chemical cleaning for fully or partially opened exhaust gas heat exchangers must be scheduled during plant downtimes.

Water Injection

Water injection is a mechanical process that was first implemented in combined heat and power stations that have diesel engines as refitting and retrofitting measures. It is based on the soot blower effect familiar from steam boiler operation. One example of a successful installation is the peak load diesel powered CHPS with 2 × 1,000 kW_{el} in operation in Lülsfeld since 1979. By 1991, it had been in operation 18,000 hours and there has been good experience with injection cycles of approximately 10 minutes with spray times of only 5–6 s.

However, this method cannot be considered a universally applicable and reliable measure. Water injection must be applied periodically and increases adverse impacts on the environment. In addition, condensate that contains soot must be disposed of.

Scrubbing with Exhaust Gas Condensate

Cooling exhaust gases to very low temperatures, e.g. a coolant temperature of merely 15°C, causes a large quantity of condensate to separate, which is able to wash out the deposits. The low coolant temperatures required limit this measure's effectiveness to the low temperature range of a heat exchanger.

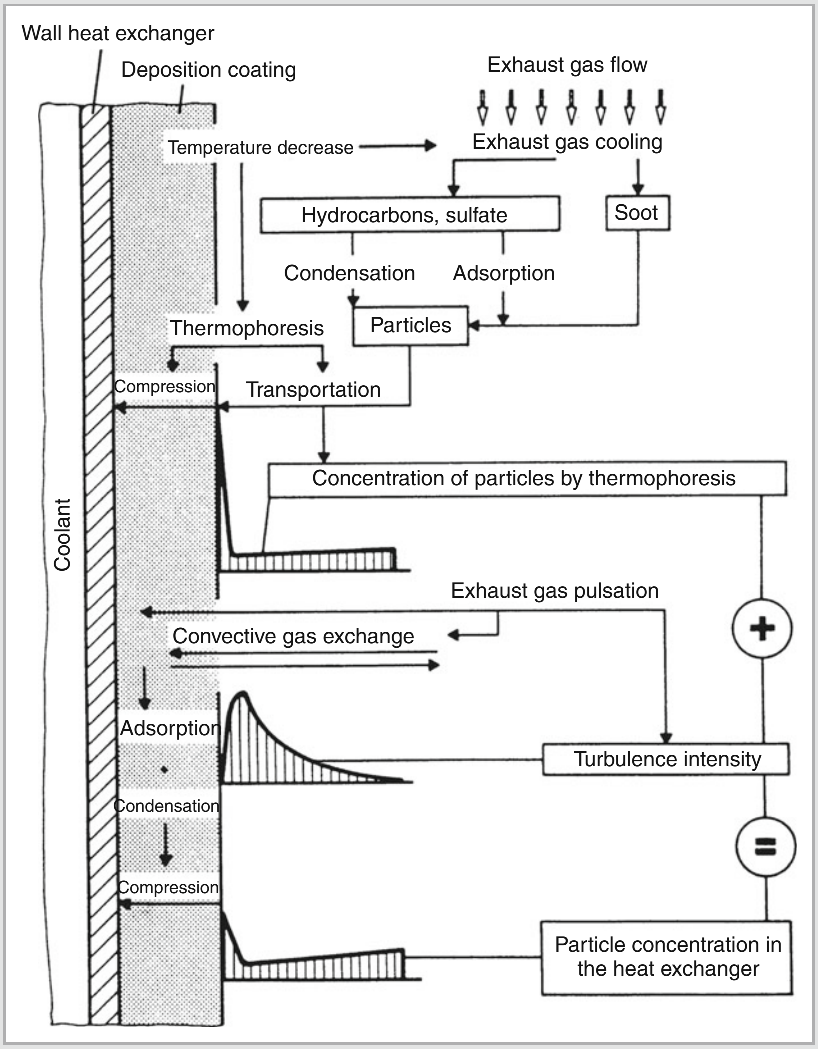

Fig. 9-46
Deposit formation process in exhaust gas heat exchangers

Mechanical Cleaning

Shot cleaning equipment [9-77] familiar from boiler manufacturing has been implemented effectively in nearly forty large diesel heating power stations, especially for dry deposits. Roughly every 60–100 min but sometimes only just once a week, soft iron shot with a diameter of 3 mm is flung against the heating surfaces and strikes the dirt particles that have deposited there. The shot is recollected and returned to the heat exchanger with a driving air flow. Soot and dirt particulates are discharged with the exhaust flow. The increased technical complexity makes it only worthwhile for large plants though.

Thermal Regeneration of the Heat Exchanger

Implemented for heating power stations with diesel engines, [9-74, 9-75, 9-78], this method employs two identically constructed exhaust gas heat exchangers configured in a series in which a valve can redirect the exhaust gas flow. The cooling water flow can be shut off in both parts as Fig. 9-47 illustrates for the left part of the equipment filled with exhaust gas. The residual water contained in it evaporates, reaches a steam separator and then a holding tank. The dried particles detach from the walls heated by the exhaust gas and are partly separated by the cyclone installed between the two heat exchangers or reach the atmosphere with the exhaust gas. If

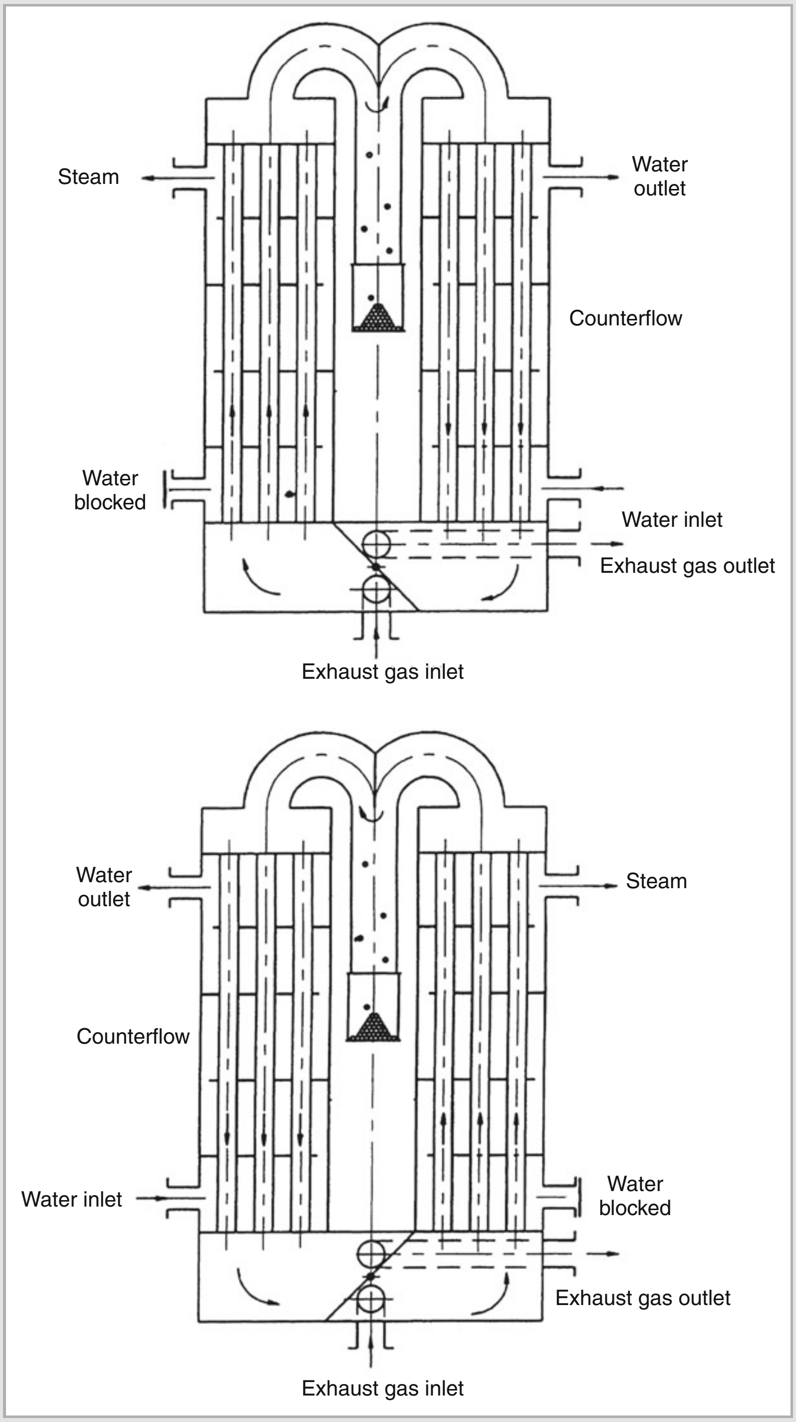

Fig. 9-47
Self-cleaning exhaust gas heat exchanger by thermal regeneration

the exhaust back pressure in the right hand heat exchanger that serves to heat the cooling water increases, the exhaust gas valve is reversed, the right hand water inlet blocked and the left hand heat exchanger reopened and the process repeats for the redirected flow of exhaust.

Turbulence Boost

Boosting the flow turbulence in the exhaust flow in the proximity of the heat transferring wall enables particles surging to the wall to return to the main exhaust flow as a result of the thermophoresis. In addition to the given exhaust pressure pulsations, pulsations generated by Helmholtz resonators may also be employed [9-75]. The propensity to form deposits turns out to diminish as pulsation amplitudes increase and as the temperature of the cooling medium increases, while the pulsation frequency apparently has no influence. These findings have still not been applied industrially though.

Exhaust gas heat exchangers (coolers) have been developed for cooled exhaust gas recirculation to meet the Euro 4 emission standard for vehicle diesel engines [9-79] (see Sect. 15.4). Built-in turbulence generating components, so-called "winglets", are mounted on the surface of the tubes to their reduce fouling [9-80]. They generate exhaust gas turbulence at acceptable pressure losses and thus reduce the propensity to foul and simultaneously improve the heat transfer. Winglets are welded on or pressure formed (Fig. 9-48). The water spaces are arranged around the exhaust ports. The port matrix is laser welded from thin high-grade steel plate (Fig. 9-49). The heat exchanger is fabricated in lengths of 100–800 mm.

An exhaust gas recirculation or EGR cooler is usually implemented in a branch line that runs from the engine's exhaust manifold to the intake manifold back to the intercooler. The flow is maintained by the exhaust gas excess pressure that is slightly above the suction pressure and occasionally by an evacuating venturi tube in the intake system as well as a non-return valve to take advantage of brief pressure waves. The exhaust flow is cooled to 100–200°C to effectively reduce NO_X emissions. The EGR cooler is connected to the coolant circuit. Apart from the high pressure application, which recirculates the exhaust to the high pressure side, concepts are being tested in which the exhaust is first extracted downstream from the exhaust turbine and downstream from the particulate filter. The exhaust mixed with the fresh air is inducted by the compressor and conducted compressed through the intercooler. The advantage of this solution is its elimination of the limit imposed on the recirculation rate by the differential pressure between the engine's exhaust manifold and inlet connection. Since the exhaust gas pressure in low pressure exhaust gas recirculation is only slightly above the ambient pressure, the EGR cooler on the exhaust side must have a significantly lower pressure loss.

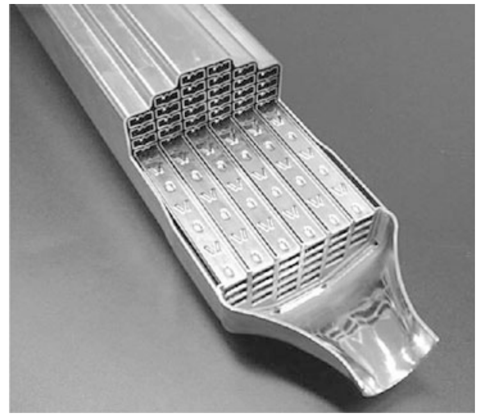

Fig. 9-49 Cross section through the tube matrix of an EGR cooler (source:Behr)

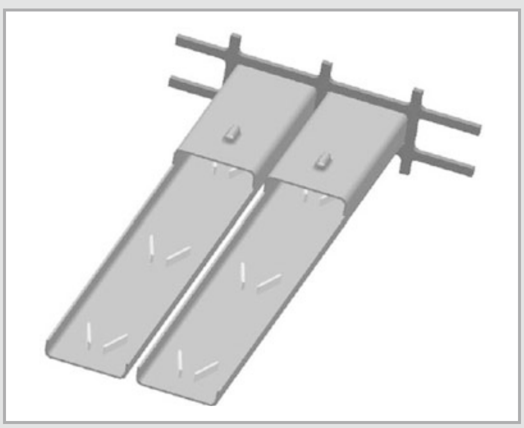

Fig. 9-48 Exhaust flow channels with winglets and tube plate (source: Behr)

High Temperature Operation

The operation of exhaust gas heat exchangers at high temperatures, i.e. >240°C, prevents components condensing from the exhaust from bonding. Such heat exchangers are implemented with thermal oil as the cooling medium or with pressurized water to generate steam. This equipment functions satisfactorily because the deposits and erosion are balanced in a deposit layer that remains approximately constant. Initially, they were usually designed in marine systems as two-pass boilers with counter flow (superheaters) and parallel flow (evaporators). While they experience fewer difficulties with fouling, they only utilize the high temperature fraction of the exhaust heat.

In principle, the use of low sulfur fuels reduces the deposition of sulfates and the formation of particulates (see Sect. 4.3). Exhaust gas heat exchangers should be designed so that the gas passages are easily accessible for cleaning purposes.

9.2.6 Coolant

9.2.6.1 Cooling Water: Properties and Requirements

Liquid cooling predominantly employs water with refining additives (engine coolant, chemicals, anticorrosion oil, etc.). The use of fuel to cool injector nozzles and sodium to cool the insides of valve stems are special cases.

An ideal coolant because of its thermal properties, water also has disadvantages with regard to engine operation:

– The freezing and boiling point limit the maximum usable temperature range to 100 K. However, nucleate boiling can intensively locally cool highly thermally loaded components (see Sect. 9.1.2 on thermal self-protection).
– Substances dissolved in water can act corrosively and/or interfere with the heat transfer through deposits.
– Malfunctions and damage in the cooling water system can cause severe damage. 15 to 20% of all engine damage can be traced back directly or indirectly to malfunctions and defects in the cooling water system.

Tables 9-9 and 9-10 compare the most important physical properties for heat exchange.

Like fuel and lubricants, cooling water is an *automotive fluid*. An engine's reliability and service life depend on its composition. Its suitability as a coolant is determined by the qualities described below.

Water Hardness

Hardness indicates the water's content of calcium ions and magnesium ions (DIN 38409 Part 6). The total alkaline earths are referred to as the total hardness, which consists of:

– Temporary hardness (carbonate hardness): Carbonates separate out of hydrogen carbonates as the temperature increases during warming and deposit as hardness scale primarily in hot spots where they inhibit heat exchange with the coolant. Since the carbon dioxide released acts corrosively, only a small percentage of carbonate hardness ought to be present when it forms.
– Permanent hardness (noncarbonate hardness): The content of calcium and magnesium chlorides and sulfates does not change with the temperature. However, it influences electrical conductivity and thus facilitates corrosion.

Water hardness is specified as the concentration of quantities of hardness ions in mmol/l. However, outmoded specifications in mval/l or German hardness (°d) are common in engine engineering. Table 9-11 contains the correlation between the quantity of dissolved hardness constituents and degrees of hardness [9-81].

Table 9-9 Physical properties of the heat transfer mediums important for engine cooling (at 80 °C and 1 bar)

Variable		Unit	Air	Water	Engine oil	Coolant[a]
Density	ρ	kg/m^3	0.986	972.0	843.6	1035
Spec. heat capacity	c_p	kJ/(kg · K)	1.010	4.194	2.154	3.59
Kinematic viscosity	ν	10^{-6} m^2/s	21.2	0.366	23.34	1.01
Thermal conductivity	λ	10^{-3} · W/(m · K)	29.93	666.6	127	429
Prandtl number	*Pr*	–	0.70	2.24	333.1	8.73

[a] Coolant: Mixture of 50% water by volume + 50 % glycol based engine coolant by volume.

Table 9-10 Physical properties of heat transfer mediums relative to air (at 80°C and 1 bar)

Physical variables of a heat transfer medium	Air	Water	Engine oil	Coolant (see Table 9-2)
Density	1	986	856	1,050
Specific heat capacity	1	4.15	2.13	3.55
Kinematic viscosity	1	0.017	1.10	0.048
Thermal conductivity	1	22.27	4.24	14.33
Prandtl number	1	3.20	475.9	12.47

Table 9-11 Overview of degrees of hardness

Ranking	Total hardness [mmol/l]	German degrees of hardness [°d]	Former German classification [°dH]	Parts per million (USA) [ppm]
Very soft	0 – 1	0 – 5.6	0 – 4	0 – 70
Soft	1 – 2	5.6 – 11.2	4 – 8	70 – 140
Moderately hard	2 – 3	11.2 – 16.8	8 – 18	140 – 320
Hard	3 – 4	16.8 – 22.4	18 – 30	320 – 530
Very hard	> 4	>22.4	> 30	> 530

Table 9-12 Required properties of engine cooling water

Total hardness [°d]	pH value [–]	Chloride [mg/l]	Sulfate [mg/l]	Hydrogen carbonate [mg/l]
max. 20	6.5. . .8.5	max. 100	max. 100	max. 100

pH Value

The pH value is a measure of the hydrogen ion concentration. Neutral solutions have a pH value of 7. Basic solutions have a higher pH value (> 7), acidic solutions a lower pH value (< 7). Engine cooling water should have a pH value between 6.5 and 8.5 (see Table 9-12).

Chloride Content

Chlorides strongly facilitate corrosion (particularly in aluminum materials and high grade steel radiators) and sludge formation. Thus, the chloride content should be minimal or less than 100 mg/l (see Table 9-12).

Corrosion, Cavitation and Erosion

Corrosion of materials and their resultant disintegration is caused by an electrochemical reaction between metals as galvanic elements and cooling water as an electrolyte. Gases dissolved in it, e.g. oxygen, carbon and sulfur dioxide, and pH values that diverge from specified recommended values intensify corrosion [9-82]. The same applies to the occurrence of flow or vibration cavitation excited by piston contact change. Vapor lock that occurs when surfaces are overheated is a special case, which, comparable to the process during cavitation, damages the material surface and its protective layers by imploding near the wall. This is hot corrosion. Erosion is to a heat exchanger what cavitation is to an engine cooling water tank. Erosion, the material attrition caused by mechanical friction between the coolant and the surface of material, is dependent on the coolant's flow velocity and the solids and gases contained in it.

9.2.6.2 The Influence of Cooling Water Routing

The danger of engine damage up through complete failure caused by unsuitable cooling water makes its preparation and care essential. Design measures can additionally decrease the potential hazards [9-81]. These include:
– air bleeding and venting the cooling circuit,
– designing lines and cross sections to facilitate the flow,
– increasing the system pressure in the cooling circuit and
– selecting optimal flow velocities.

A cooling circuit can be operated with coolant temperatures of up to approximately 115°C, an excess system pressure of up to 2 bar being necessary. The system is protected by a pressure relief valve.

Low flow velocities facilitate the formation of deposits and increase material attrition induced by corrosion depending on the material. Recommended values are 0.2 m/s<c_{Grenz}<3.0 m/s for aluminum, c_{Grenz}<2.2 m/s for CuZn20Al for sea water cooled components, <2.7 m/s for CuNi10Fe and <4.5 m/s for CuNi30Fe. The electrochemical compatibility of materials is important when coolants conduct electricity well. This holds particularly true when one cooling system is designed by different enterprises (engine manufacturers, shipyard, etc.). Finally, cooling water losses have to be prevented or minimized because refilling allows additional oxygen and carbon dioxide to reach the cooling circuit. Moreover, active ingredients can concentrate (*accumulation of minerals*).

9.2.6.3 Cooling Water Care

Cooling Water Requirements

Cooling water care begins with the selection of the water. Engine manufacturers specify clean, clear water free of

impurities. However, condensate or completely deionized water may also be employed. Sea water, brackish or river water and rainwater are fundamentally unsuitable. Soft water prevents scale deposits (see Table 9-11), yet must have a minimum hardness of 2°d to prevent increased metal ion solubility (see Table 9-12).

Engine Coolant

Antifreezes

An engine coolant must be added to the engine cooling water when an engine is operated at ambient temperatures at or below the freezing point,. Its basis is *ethylene glycol* (ethane-1,2-diol) or less often *propylene glycol* (propane-1,2-diol). Its effectiveness depends on the blending ratio with the water. A glycol content of approximately 50% by volume lowers the mixture's freezing point to approximately −50°C. The effect diminishes again when percentages are greater (Fig. 9-50).

The percentage of glycol also changes a coolant's physical properties (density, heat capacity, viscosity, thermal conductivity and boiling point). This must be factored into radiator design (Fig. 9-51) [9-53].

Since a water/glycol blend provides insufficient protection against corrosion, anticorrosion agents (*inhibitors*) are additionally blended into commercial engine coolants.

Siliceous Engine Coolants

Siliceous engine coolants predominantly contain inorganic inhibitors, e.g. silicates, nitrites, nitrates or molybdates, as well as organic inhibitors, e.g. tolyl or, to a small extent, benzotriazole. Additives, e.g. borate, phosphate, benzoate or imidazole, serve as buffer substances to assure the desired, mildly alkaline pH value throughout the entire period of operation. In addition, detergents (e.g. sulfonates), antifoaming agents and pigments are also added.

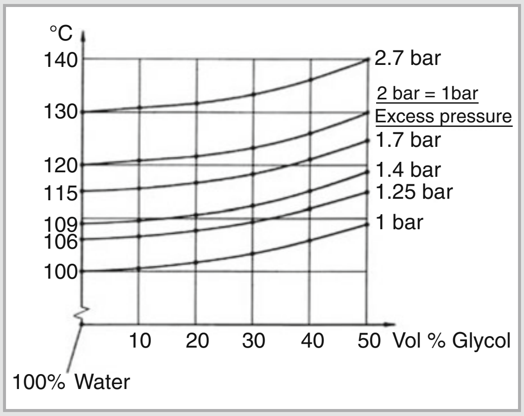

Fig. 9-51 Boiling point curves of water/ethylene glycol mixtures as a function of pressure

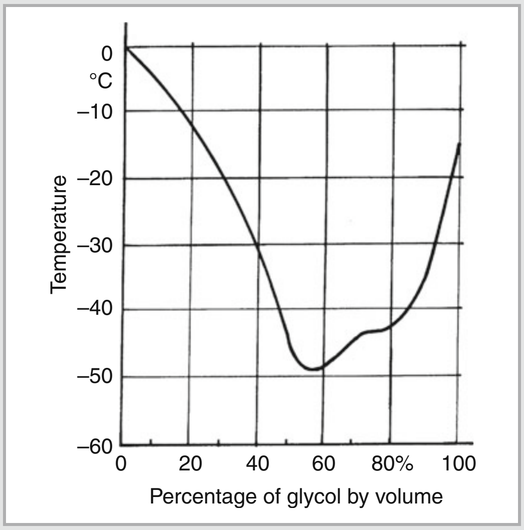

Fig. 9-50 Change in the freezing point of water/ethylene glycol as a function of glycol content by volume

These engine coolants have proven themselves in the field over decades. Their silicate content counteracts dangerous hot corrosion in aluminum engines by forming thin protective coats. However, the low solubility of some additives is disadvantageous. Thus the concentration of inhibitors cannot be increased randomly. Inorganic additives also degrade in normal driving, thus making a new dosage or replacement necessary if the minimum concentration is fallen below.

OAT (Organic Acid Technology) Engine Coolant

OAT engine coolants have been available for several years and contain a combination of organic rather than inorganic inhibitors. Combinations of aliphatic monocarboxylic and dicarboxylic acids, azelaic and aromatic carboxylic acid are typical. Blends of carboxylic acid with inorganic inhibitors are also employed but silicates are usually never employed. The advantage of OAT products is their slower degradation in normal driving. This increases their working life and thus reduces the maintenance required.

Chemicals

Chemicals are only used when there is no danger of freezing and engine manufacturers' specifications allow them. Additives used do not contain any glycol and thus only protect against corrosion. However, this increases evaporation losses and thus the maintenance intervals. The danger of corrosion

also increases when there are mechanical loads. Chemicals are usually added in a concentration of 5–10%.

Anticorrosion Oils

Infrequently employed, anticorrosion oils' protection against corrosion and cavitation is based on their prevention of oxygen and other gases dissolved in the cooling water from reaching the walls in engine cooling water chambers by forming a protective film and on their stoppage of electrochemical corrosion. Added in a blending ratio of 1:200 to 1:100 (70), anticorrosion oils consist of emulsifiable mineral oils with additives to protect against corrosion or sludge deposits should the emulsion break, e.g. when heated to $>95^{\circ}C$ or in contact with copper materials.

A general rule, engine manufacturers' specifications as well as compatibility for humans and the environment during use and disposal have to be observed for every additive.

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